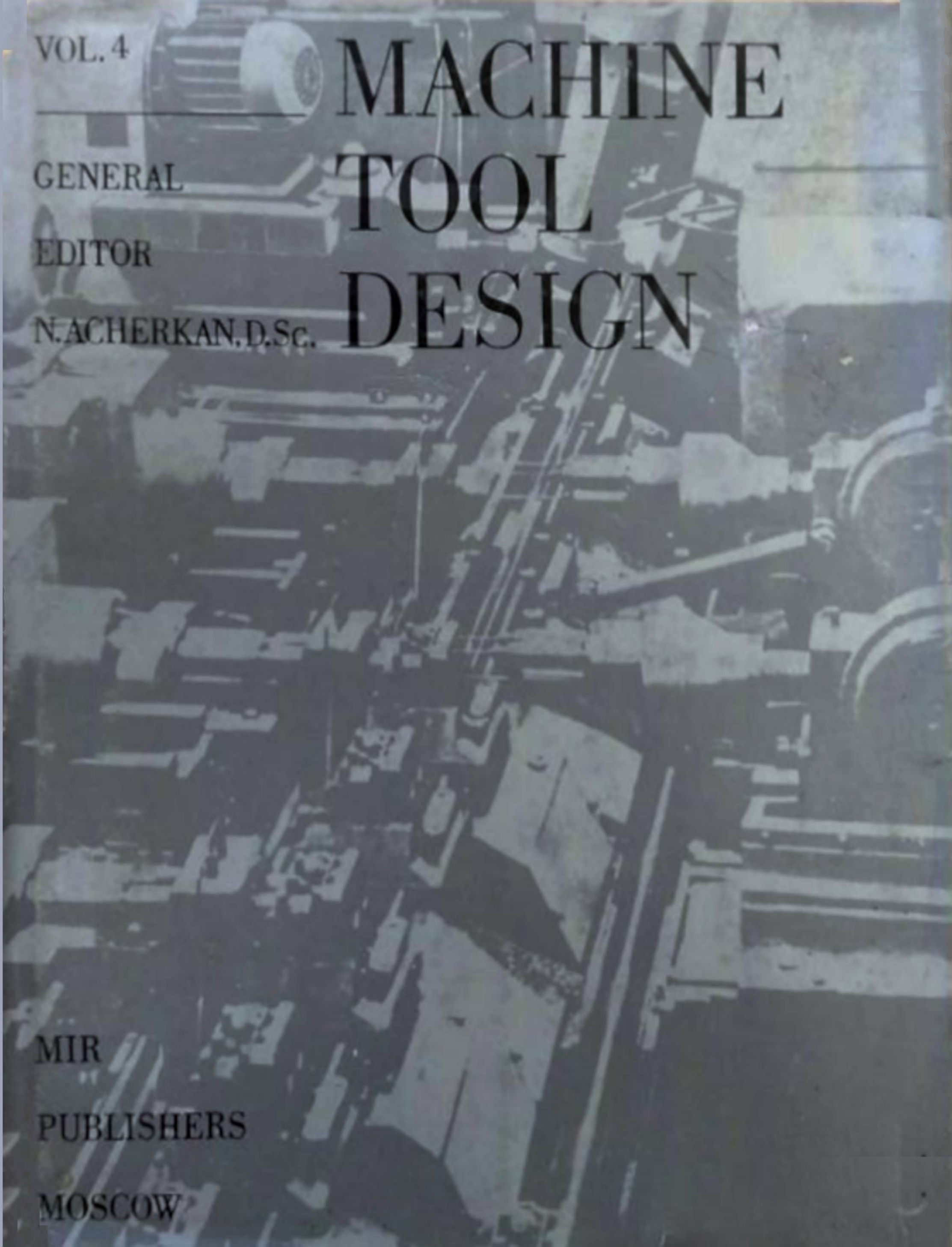
VOL. 4

GENERAL EDITOR N. ACHERKAN, D.Sc.

MACHINE TOOL DESIGN



MIR PUBLISHERS MOSCOW

This fundamental four-volume work was written by scientists and specialists on the teaching staff of the Machine Tool Engineering Institute in Moscow. The editor, Prof. N. Acherkan, D.Sc. (Eng.), is an eminent Soviet specialist in machine tool design and the author of over fifty scientific works and textbooks in this field. Prof. Acherkan holds the title of Honoured Scientist of the Russian Federation and has occupied the chair of Machine Tool Design for 35 years.

Prof. Acherkan graduated from the Warsaw university in 1914, where he majored in mathematics. He received a gold medal for work done in the university. In 1920 he received a second degree from the mechanical engineering department of the Petrograd Polytechnical Institute.

The present work was translated from the considerably revised and recently published second edition which includes all the latest achievements in world machine tool practice. The earlier edition has become an indispensable handbook for Soviet designers. Therefore, the publishers feel that this work should be of great value to engineers engaged in the design, manufacture and maintenance of machine tool equipment. It can also be used to advantage by the students of engineering institutes majoring in Process Engineering, Metal-Cutting Machine Tools or Cutting Tool Design.

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PUBLISHERS

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МЕТАЛЛОРЕЖУЩИЕ СТАНКИ Том 4

ПОД РЕДАКЦИЕЙ
Засл. деятеля науки и техники РСФСР
д-ра техн. на к проф. Н. С. АЧЕРКАНА

ИЗДАТЕЛЬСТВО «МАШИНОСТРОЕНИЕ» МОСКВА

MACHINE TOOL DESIGN

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TRANSLATED FROM THE RUSSIAN BY NICHOLAS WEINSTEIN

Vol. 4

MIR PUBLISHERS

MOSCOW 1969

UDC 621.90(022) = 20

Revised from the 1965 Russian edition

Russian Alphabet and Transliteration

А а	a	П п	p
Б б	b	Р р	r
В в	v	С с	s
Г г	g	Т т	t
Д д	d	У у	u
Е е	e	Ф ф	f
Ж ж	zh	Х х	kh
З з	z	Ц ц	ts
И и	i	Ч ч	ch
К к	k	Ш ш	sh
Л л	l	Щ щ	shch
М м	m	Э э	e
Н н	n	Ю ю	yu
О о	o	Я я	ya

На английском языке

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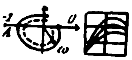


MACHINE TOOL TESTING AND RESEARCH

PART SIX

AUTOMATIC MACHINE TOOLS AND TRANSFER MACHINES

CHAPTER 1
INTRODUCTION

1-1. Significance and Development of Machine Tool Automation

The automation of a machine tool increases the productivity of the operator's labour due to the increase in the production capacity of the machine tool. The operator is freed of direct participation in the machining process and it becomes feasible for him to operate several machine tools at the same time.

In the machining of metals (and other materials), integrated automation of the manufacturing process is accomplished by incorporating machine tools into an automatic transfer machine or line.

Automation reduces the physical effort required of the operator, frees him of tediously repeated movements and from monotonous nervous and physical stresses. At the same time, automation requires a much higher level of servicing, both in setting up the machine and in subsequent regular operation. It lightens the burden of physical labour of the operator by increasing the share of required brain work.

As a result of the increase in production capacity in automation, the required number of machine tools is reduced and a higher output is achieved per unit of shop floor space. Automation imparts a rhythmical pace to the machining process and ensures stable quality of the blanks and workpieces in all stages of manufacture.

The devising of a drive mechanism for the uniform rotation of clock hands led to the development of drive mechanics which was utilized later in building processing machinery.

Water wheels of flour mills were used to power processing machinery, including metal-cutting machine tools.

In Russia metal-cutting machine tools, operating on an automatic cycle, first appeared at the beginning of the 18th century in the production of armament.

A soldier of the Oranienburg battalion, Yakov Batishchev, sent to the Tula Small Arms Works by order of the Moscow Ordnance Office, equipped a "barn" for the production of musket barrels with new machine tools that were ready for operation by January 1715. Two of these machines, intended for external "rubbing" (finishing) of the barrels with files and for internal "reaming" of the muzzle, are illustrated in Figs. 1 and 2.

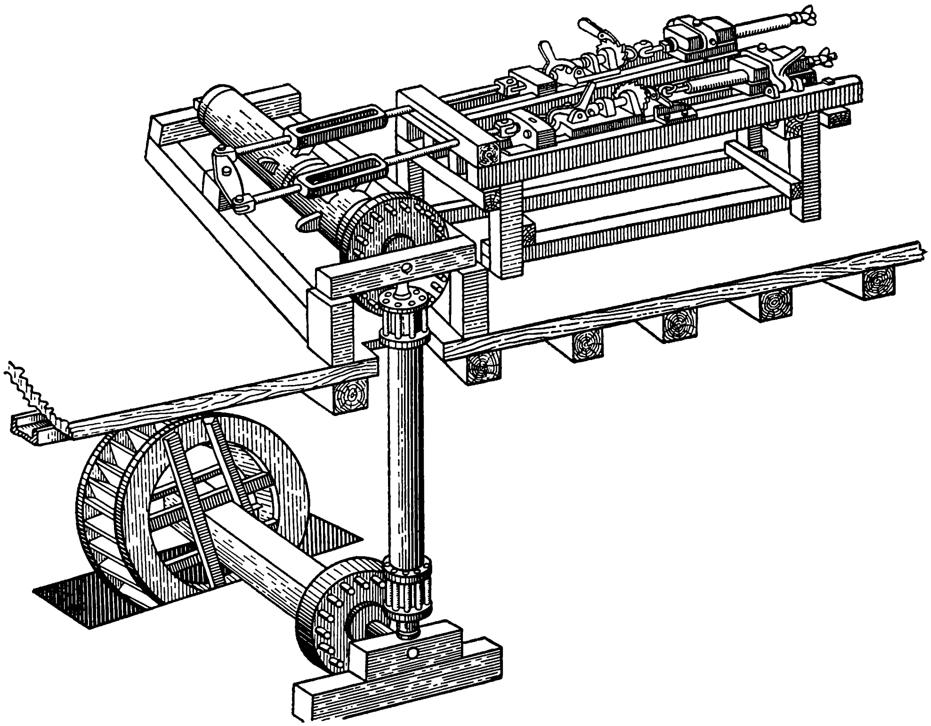


Fig. 1. A part of the "reaming" machine devised and built by **Yakov Batishchev in 1715**

For such different operations as the external and internal finish machining of musket barrels, Batishchev applied a manufacturing technique with the same kinematic relative motions of the tool and workpiece, enabling him to unify the two machines to the maximum possible extent.

In "whitening", the barrel was reciprocated between two 30-pound files with teeth cut on the concave semicylindrical surface. After each stroke, the barrel had a periodic circular feed motion. In "reaming", the barrel was clamped and the "reamer", a sort of internal file, was reciprocated and had a circular feed motion after each pass or stroke.

In this layout, an overshot water wheel (Fig. 3) drove a horizontal wooden shaft which, through a vertical shaft and two sets of pin gearing, transmitted rotation to a horizontal shaft on the upper floor. Secured on this shaft were iron "cranks" (actually lugs) in line with the longitudinal axes of the twin-mounted machines. In their rotation, the cranks entered the slots of ties which, through pivots, linked the wooden carriages with a rocker

arm. It was necessary to link together two tie rods with a rocker arm since the cranks entering the tie rod slots could move the carriages only in one direction. The return stroke of each carriage was effected by the other tie rod through the rocker arm.

Mounted on the carriage was the "darting ratchet", i.e., a ratchet wheel mechanism, tripped on each stroke of the carriage by a wedge-shaped dog which was secured on the bed and turned a lever carrying the ratchet pawl. The barrel was clamped to the ratchet wheel shaft in "whitening" or to the "reamer" in finishing the muzzle bore.

Hence, these machines operated on a semiautomatic cycle. The operator unloaded and loaded the workpiece and adjusted the reamer and files (setting up the machine). These were the first semiautomatic machine tools in the history of machine tool engineering.

Pavel Zakhava, a navy mechanic by training, reconstructed the Tula Small Arms Works in 1810. He introduced interchangeable manufacture of the components of the cocking piece mechanism, and was the first to use

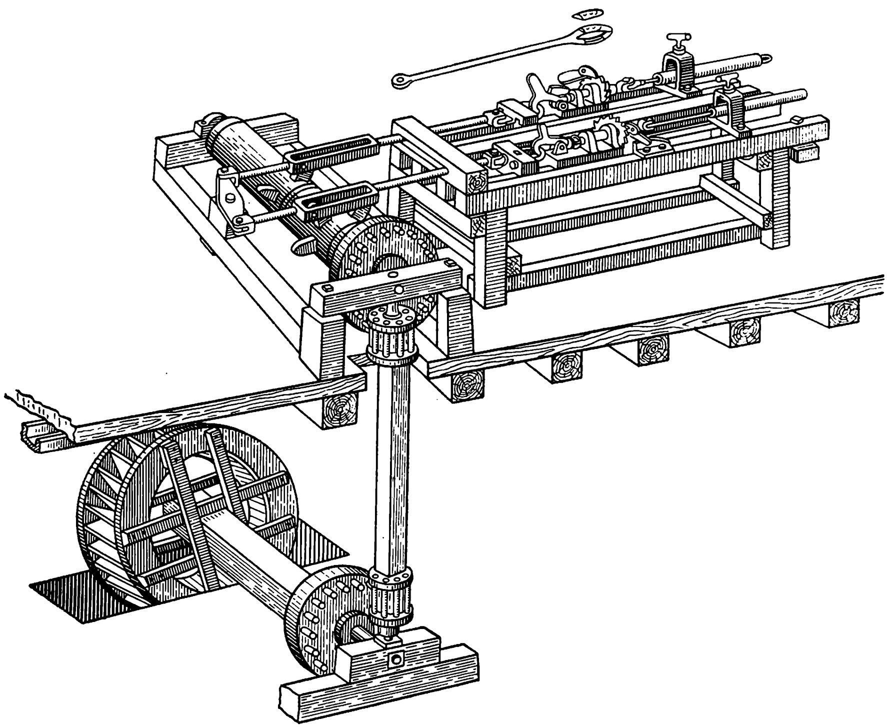


Fig. 2. A part of the "rubbing" machine devised and built by Yakov Batishchev in 1715

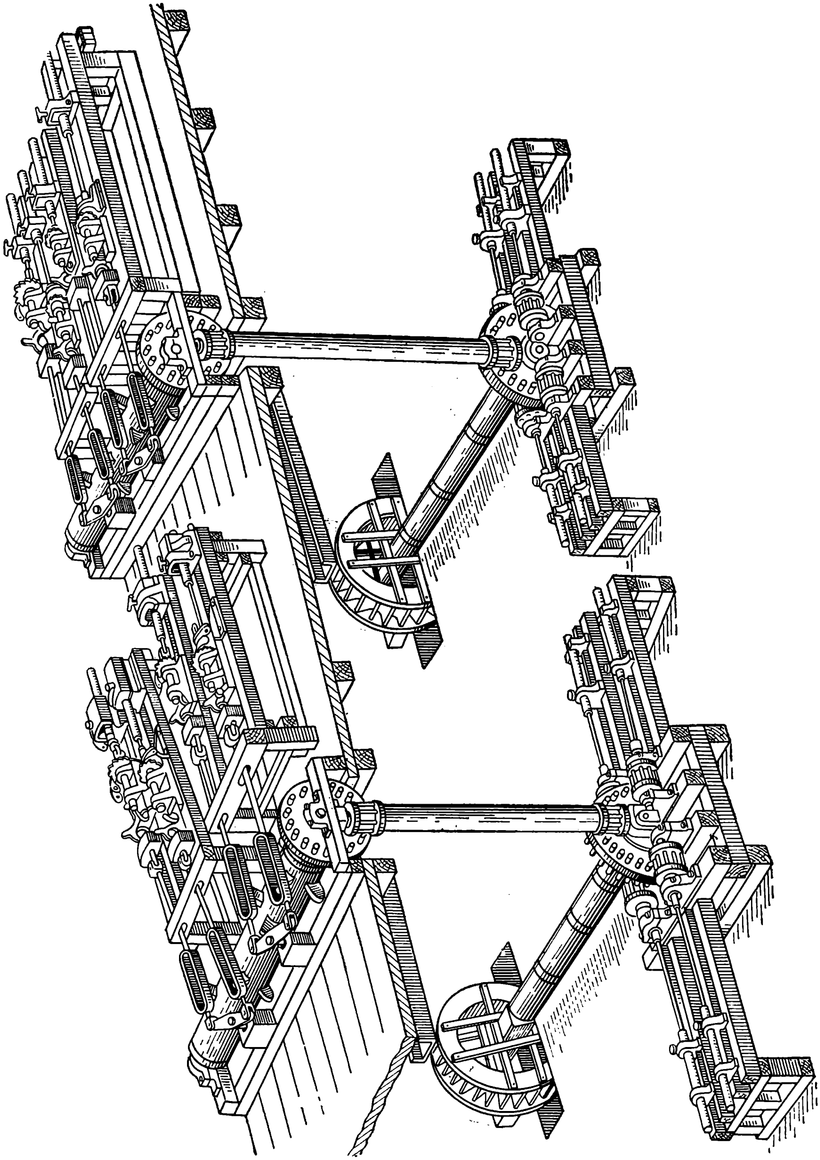


Fig. 3. Layout of the water-powered equipment developed by Yakov Batishchev at the Tula Small Arms Works in 1715

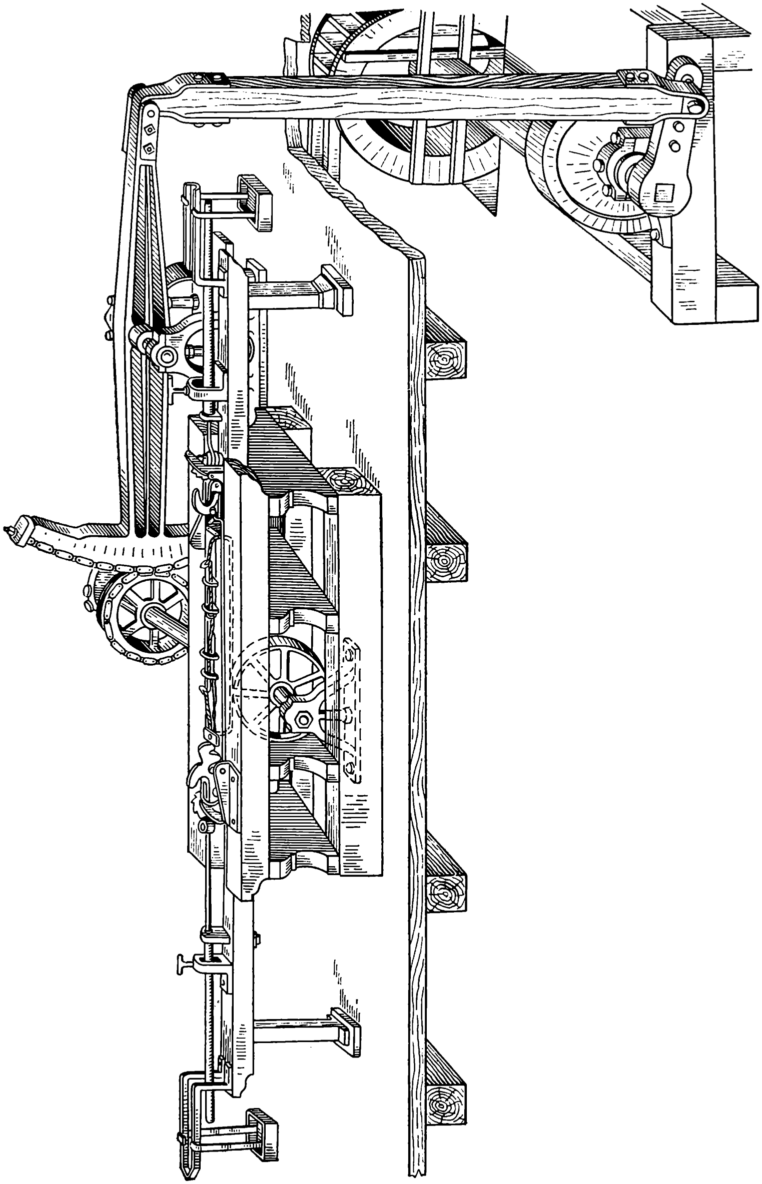


Fig. 4. A part of the "reaming" machine developed by Pavel Zakhava in 1810

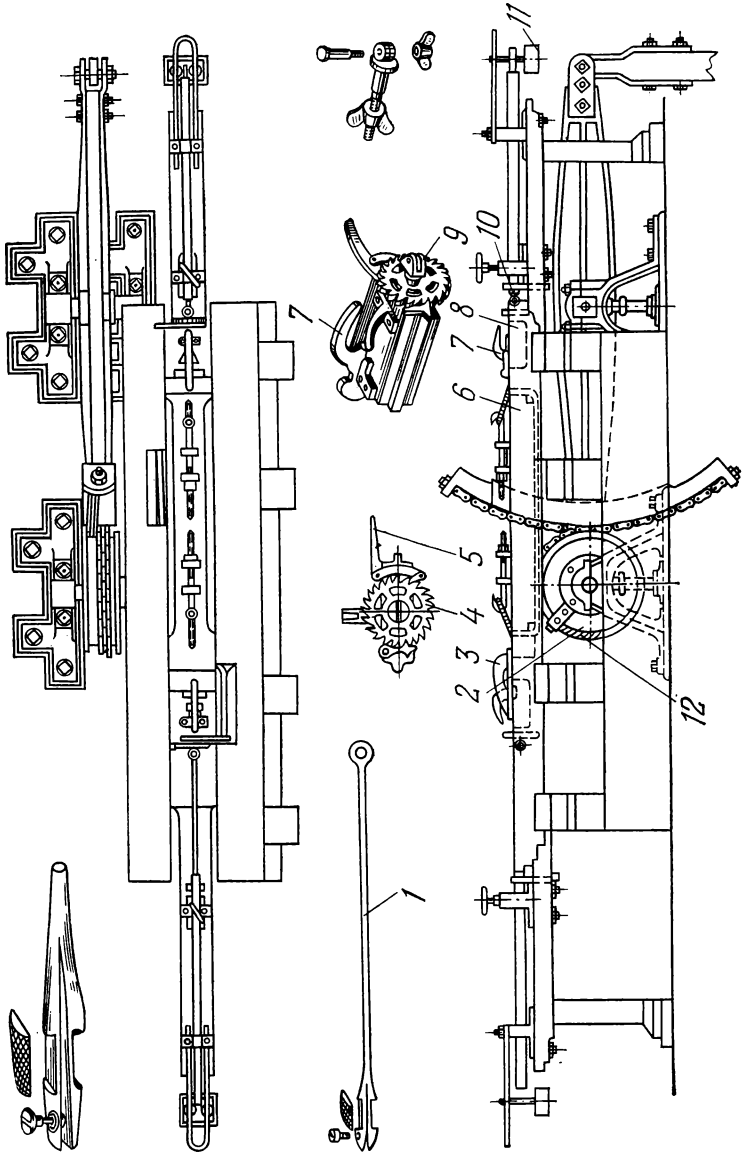


Fig. 5. Diagrams of units and parts of Zakhava's "reaming" machine

a "milling stone", i.e., milling cutter (milling was first used in England in 1854).

Pavel Zakhava redesigned Batishchev's reaming machine, using a crank gear for driving the input shaft (Fig. 4). In the new design the crank radius was adjustable, enabling the stroke of the carriages to be changed. Linked to the upper end of the connecting rod was a rocker arm with a sector which, by means of a chain, imparted a reversing rotary motion to the iron input shaft. Pulleys *2* (Fig. 5), mounted on this shaft, transmitted a reciprocating motion to slide *6* by means of two ropes *12*. Linked to the ropes by means of hooks *7* were carriages *8* with the ratchet mechanism *4*. Eye *10* of "reamer" *1* fitted into slot *9* of the ratchet wheel to which it was linked with a pin. Lever *5* ran onto wedge-shaped bar *3*, thereby rocking the pawl and transmitting a periodic circular feed to the ream. Fine chips, formed in reaming, were accumulated in sheet metal box *11*.

Zakhava redesigned Batishchev's "rubbing" machine in a similar manner, using a greater number of cast-iron parts than in the reaming machine.

In 1824 Pavel Zakhava built a lathe with a copying attachment, follower rest and automatic feed disengagement of the carriage at the end of the cut. In the turning operation, the musket barrel was mounted on a steel bar, stretched like a wire between the stocks, and was supported by a rest.

Another trend in machine tool automation, originating from clock works and associated with the development of drive mechanics, was represented in the activities of Andrei Nartov, first a student and later a teacher of the School of Mathematical and Navigational Sciences which was founded by Peter the Great and was located in the Sukharev Tower in Moscow.

In 1712, Nartov built the first copying lathe with a power-feed carriage (Fig. 6). The cutting carriage *6*, carrying the single-point tool, and the copying carriage *1*, carrying the stylus, had different rates of power longitudinal feed. This enabled the profile of workpiece *5* to be turned with a steep slope to a template *3* having a profile with a gentle slope. Cross feed of the tool, effected by template *3* mounted on spindle *2*, was accomplished by transverse swivel of upright *4* about pivot pin *7*. Upright *4* carried the front bearing of spindle *2*; the rear end of the spindle was supported by centre *8*. The lathe drive was powered by hand; the lathe was intended for precise finishing operations. Nartov also built a copying lathe for the transverse turning of medals (medallion lathe).

After building a thread-cutting lathe for the manufacture of screw presses, Nartov built a copying lathe for the Moscow mint in 1728. It had differential feeds of the cutting and copying carriages along a lead screw.

The 19th century saw priority development of the light industries—textile and food—in Russia. In the field of machinery, thermal power plant equipment was in general demand in the factories of the light industries. Since Russia of that time was a country using expensive fuel, the engineering

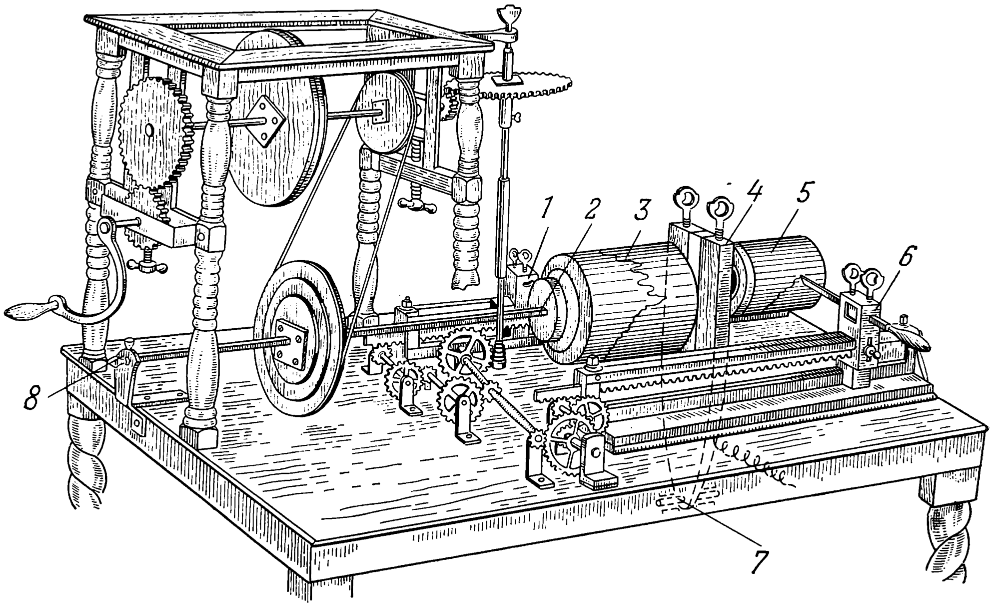


Fig. 6. Kinematic diagram of a copying lathe devised and built by Andrei Nartov in 1712

requirements made to heat engines were very high, and Russian plants competed with success with foreign companies in the production of steam and internal-combustion engines. Such plants included Bromley Bros. of Moscow, the Sormovo Plant, the Kolomna Plant, Nobel of St. Petersburg and Felzer of Riga.

The light industries required metal-cutting machine tools for repair shops, in small amounts but in a wide variety. In addition, the duty on imported machine tools was low in comparison with other machinery, since it was figured on a per pood* basis without taking into consideration the fact that a machine tool has a much greater number of machined and fitted parts. Under such conditions, Russian engineering plants could not compete with foreign companies in the production of machine tools.

To protect native industry, the tsarist government forbid state plants to import equipment from abroad if offers were made by Russian plants to supply this equipment. This maintained machine tool production at the Bromley Bros. Plant in Moscow (now the Krasny Proletary Plant), which supplied railway repair shops with lathes, and the Phoenix Works in St. Peters-

* 1 pood = 16.38 kg.

TABLE 1

New models	Years							
	1932	1937	1940	1945	1950	1955	1957	1965
Total, including	47	190	320	150	384	788	900	1500
automatic machine tools	7	42	87	40	115	250	295	650

burg (now the Sverdlov Plant), which manufactured plate planers and radial drills with a large drilling radius for the Navy Department.

No automated machine tools were developed in the 19th and 20th centuries in prerevolutionary Russia.

A single-spindle automatic screw machine with a magazine was developed by Spencer in the USA in 1873, while the bar-type automatic with a turret was developed by Worsly in 1880. Multiple-spindle automatics first appeared in the USA in the nineties of last century and in Europe at the beginning of the 20th century.

After the decision in December 1925 to intensify industrialization of the country, the engineering industries began to grow rapidly in the USSR, together with the machine tool industry. The growth in the production of automatic machine tools can be seen in Table 1, which lists the number of new models, in the type and size ranges, that were put into regular production in the years indicated.

Mainly general-purpose machine tools were put into production during the years of the First Five-Year Plan.

By the end of the Second Five-Year Plan the production of automatic machine tools was begun on a wide scale. In the period of the postwar Five-Year Plan (1945-1950), there was a substantial increase in the production of unit-built machine tools due to the development of automatic transfer lines and machines.

1-2. Current Objectives in the Field of Machine Tool Automation

One of the main objectives in Soviet industry is the integrated automation of manufacturing processes. Therefore, the possibility of building a machine tool into an automated production line or automatic transfer machine is one of the important requirements to be considered in designing a new automatic machine tool. For this purpose, the layout of the units must be tied in with the handling system of the transfer machine or line, as well as with its system of loading devices. The control system of the new model should incorporate

facilities for starting and stopping the machine from the control desk of the transfer machine. Provision should be made for automatic chip disposal from the workpiece machining zone and from the machine tool.

Of vital importance in integrated automation are automatic gauging and automatic feedback readjustments of the setup.

In contrast to semiautomatic machine tools, carbide-tipped cutting tools cannot always be used to their full capacity in present-day automatic machine tools. Hence, a fuller utilization of such tools in automatics is one of the currently urgent problems in machine tool engineering.

In the engineering industries, about 80 per cent of the parts are manufactured in small lots. Therefore, another vital problem is the development of automated machine tools and automatic transfer machines that can be readily changed over from one type of workpiece to another. This problem is being solved by manufacturing machine tools with numerical-control devices, by employing tracer-control devices with servosystems, by the use of quick-change tooling—interchangeable toolholders, and quick-change multiple-tool holders which are set up separately from the machine—and by the development of automatic machine tools and transfer machines that can be quickly set up to machine any one of a group, or "family", of workpieces.

1-3. Basic Concepts and Definitions

The concept of an automatic machine tool includes:

(1) the manufacturing process or sequence of machining operations which determines the motions of the working members, classified as main (directly participating in the cutting process) and auxiliary (not directly participating in the cutting process);

(2) operative mechanisms of the working members for imparting the movements of their working cycle to these members;

(3) transmitting mechanisms of the drive for the operative mechanisms of the working members;

(4) self-acting system for controlling the mechanisms of the machine tool to accomplish the working cycle of machining without the participation of the operator.

The system of automation of a machine tool should be considered as an integrated whole.

1-4. Operating Cycles of Automatic Machine Tools

An *automatic operating cycle* is the whole complex of periodically repeated movements of the main and auxiliary working members of a machine tool that are required to machine a workpiece and that take place under the con-

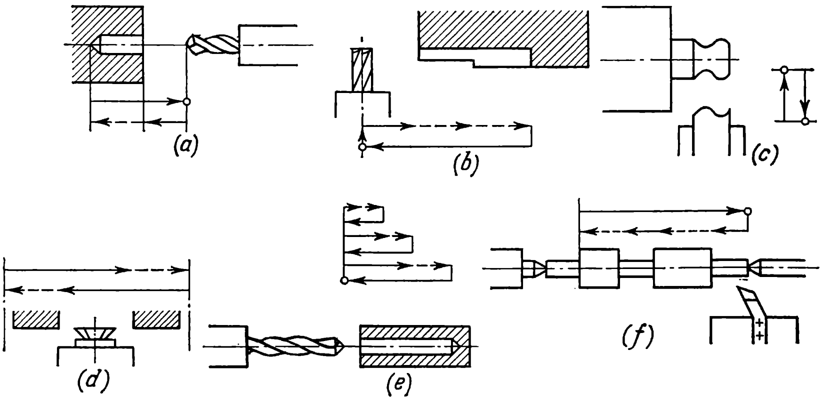


Fig. 7. Diagrams of simple cycles:

(*a*) simple cycle; (*b*) cycle with two different working feeds; (*c*) form turning with a dwell for cleaning up; (*d*) reciprocal milling cycle (a new blank is mounted while another is being machined); (*e*) repeated elements of the cycle; (*f*) intermittent-feed cycle (cycle with skip motions)

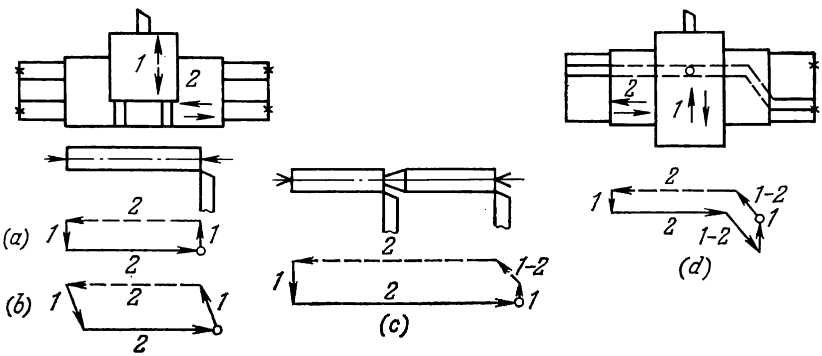


Fig. 8. Diagrams of complex cycles of the front (longitudinal) carriage of a semiautomatic lathe:

(*a*) only straight withdrawal possible; (*b*) approach and withdrawal with travel of both saddle and cross slides; (*c*) straight withdrawal and angular infeed with feed of both saddle and cross slides; (*d*) retraction of cross slide for tool relief, angular withdrawal and approach of cross slide and angular infeed to a cam

ditions of automatic control in a definite sequence, according to definite laws of motion, at definite sections of the paths.

The cycle in automatic machine tools is usually of the closed type, i.e., at the end of the cycle each working member comes to a position which is its initial point in the next cycle.

A general automatic cycle consists of the particular cycles of the main and auxiliary working members. The main working members of an automatic machine tool participate directly in the cutting process. Auxiliary members do not participate directly in the cutting process, though their operation is part of the general automatic cycle.

A *semiautomatic cycle* includes certain nonautomatic handling operations which are accomplished with the participation of the operator. As a rule, such operations include loading, clamping, unclamping, releasing and removing the workpiece, and starting the machine to repeat the operating cycle. As their name infers, *semiautomatic machine tools* operate on a semiautomatic cycle.

In the *setting-up mode of operation* of automatic and semiautomatic machine tools the movements of the working members are manually controlled by the operator (or setter-up).

1-5. Particular Cycles of the Operative Members

In their general nature, the cycles of the main and most of the auxiliary working members are closed, repeating cycles of rectilinear reciprocating motions (much less frequently—rocking motions).

The particular cycles of the main working members may be either simple or complex. Simple linear cycles are accomplished along a straight line. They consist of rapid approach (denoted by⟶), working feed (– – –→), dwell (0) and rapid withdrawal (⟵). In some cases certain of these elements are absent and in others they are repeated (Fig. 7). In complex cycles, the main working member travels in mutually perpendicular directions (Fig. 8).

Most of the auxiliary working members execute closed cycles of reciprocating motion. Certain auxiliary working members may have a rotary and, in most cases, indexing cycle of motions for changing the stations or positions of the workpieces (spindle carriers of multiple-spindle automatics, spindle carriers of vertical multiple-spindle semiautomatics and work tables of multiple-station unit-built machines) or of the tools (turrets of turret lathes, indexing attachments for drilling and tapping in single-spindle automatic screw machines, etc.).

CHAPTER 2

CAM-CONTROLLED AUTOMATIC MACHINE TOOLS

2-1. Structure of Cam-Controlled Automatic Machine Tools

Each machine tool consists of motion (M), transmitting (T) and operative (O) mechanisms and a working member (W).

A general block diagram of a metal-cutting machine tool is shown in Fig. 9.

Transmitting mechanisms have a rotary motion. The main working members and most of the auxiliary members of automatic machine tools execute closed cycles of reciprocating motions. Hence, the operative mechanisms of most working members serve to convert rotary motion into reciprocating motion. These mechanisms are further complicated by lever or other devices for transmitting motion to the working members.

In addition, certain auxiliary working members make rotary motions and, in the majority of cases, indexing motions for changing positions or stations. In these cases, the operative mechanisms are rotary mechanisms for periodic motion, operating on closed rotary cycles.

The block diagram shown in Fig. 9 covers both the main and auxiliary working members.

To enable various particular motion cycles of the working members of automatic machine tools with a mechanical drive to be accomplished without disconnecting the kinematic linkages between the various members themselves and between the members and the transmitting mechanism T, it will be necessary to incorporate a device (K_c) providing for a cyclic change in the kinematic linkage between the transmitting mechanism T and the working member W in the particular drive of each working member. Changes in the motions of the working member take place during its closed and repeated operating cycle due to this device K_c.

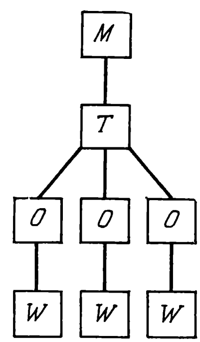


Fig. 9. Block diagram of a metal-cutting machine tool

The operative mechanisms of the working members of automatic machine tools can be divided into two groups:

(1) Operative mechanisms which incorporate K_c, a device for a cyclic change in the kinematic linkage between the transmitting mechanism T and the working member W. These are cyclic operative mechanisms O_c of which the most widely used are cam mechanisms.

(2) Operative mechanisms which do not incorporate such a device K_c for a cyclic change. These are noncyclic mechanisms O_n of which the most commonly used are the motion screw-and-nut mechanisms.

2-2. General Structural Properties of Cyclic Operative Mechanisms

Single-revolution feature of a cyclic operative mechanism. To obtain a closed repeating cycle of motions of the working member, the input, or driving, shaft of a cyclic operative mechanism makes one revolution for each cycle of motions of the working member. This single-revolution feature is characteristic of all cyclic operative mechanisms.

Self-engagement of working member motion. In the course of continuous rotation of the input shaft of the cyclic operative mechanism, the cycle of motions of the working member begins automatically at a definite stage in the rotation of the shaft.

The device for starting or engaging the cycle of motions of the working member (E_s) is incorporated in the operative mechanism itself. In this respect, the most adaptable is a cam mechanism in which the law of motion of the working member depends upon the cam profile and is arbitrary within certain limits. Other cyclic operative mechanisms (crank gear, link motion and Geneva wheel mechanisms) do not have this adaptability since the motion of the working member adheres to a definite law.

Continuity of the cyclic kinematic linkage between the input shaft of the operative mechanism and the working member. Characteristic of cyclic operative mechanisms is the continuity of the kinematic linkage between the driving shaft of the operative mechanism and the working member, not only in case of continuous motion of the latter, but also during its intermittent motion. Dwell in cam mechanisms or the idle period between consecutive movements of the working member in a link or Geneva wheel motion is a part of the regular operating cycle of this member. The beginning and end of the cycle are determined by the initial (which is also the final) position of the driving shaft of the cyclic operative mechanism.

Block diagram of an automatic machine tool with cyclic operative mechanisms. Here, the following elements, having definite structural properties, can be singled out: Sh_{in}—the driving or input shaft which makes one revolution for each cycle of the given working member, the device K_c for a cyclic change in the kinematic linkage between the driving shaft of the operative mechanism and its working member, and E_s—the device for engagement and disengagement of the cycle of motions of the working member. Such a block diagram is illustrated in Fig. 10.

All of these elements are typical of every cyclic operative mechanism. For simplicity, therefore, we shall incorporate its block diagram in the

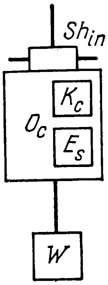


Fig. 10. Block diagram of a cyclic operative mechanism

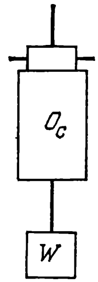


Fig. 11. Simplified block diagram of a cyclic operative mechanism

general block diagram of an automatic machine tool in its simplified form (Fig. 11).

If each working member of an automatic machine tool executes one particular cycle in the course of the automatic cycle of the machine, then all the driving shafts in the operative mechanisms of the working members will make one revolution during the cycle. If they rotate at uniform and equal speed, the driving shafts in the operative mechanisms of all the working members can be united into a single common shaft (Fig. 12) which makes one revolution during a general automatic cycle of the machine tool. This shaft is called the main camshaft. When the particular cycles of the working members are thus co-ordinated, cam-type mechanisms are employed to great-

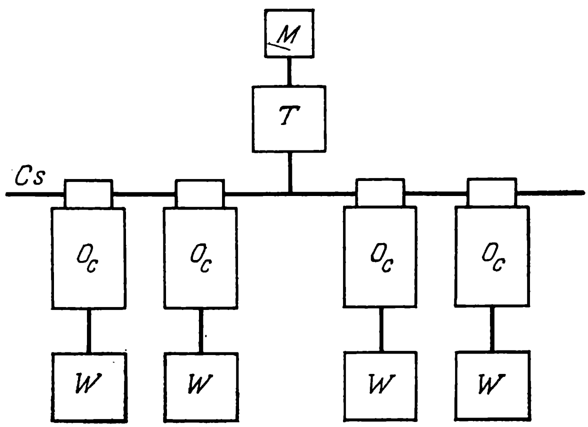


Fig. 12. General block diagram of an automatic machine tool with cyclic operative mechanisms

er advantage because they enable the law of motion of the working member to be established arbitrarily (within certain limits) in accordance with the cam profile. This makes it possible to obtain a law of motion for each working member in compliance with the nature of its operating cycle. At the same time, the beginning and end of the cycle of motions of the working member can be established with great flexibility, maintaining proper co-ordination, however, of the general automatic cycle.

Other cyclic operative mechanisms (Geneva wheel, link motion and crank gear mechanisms), in which the motions of the working member are governed by definite laws, do not possess this flexibility. For this reason, the cam mechanism has become the principal cyclic operative mechanism for the main and auxiliary working members of automatic machine tools, and automatic machine tools with cyclic operative mechanisms are always of the cam-controlled type.

The uniformly rotating crankshaft positively and continuously links all operative mechanisms O_c and ensures the co-ordination of motions of all working members during the working cycle.

2-3. General Properties of Automatic Machine Tools Having Cyclic Operative Mechanisms (Cam-Controlled Automatic Machine Tools)

Due to the positive and continuous kinematic linkage between the camshaft and working member, each element of the particular cycle of the working member can be precisely and stably limited to the minimum required displacement of the working member with the minimum required loss of time. This leads to increased output of the machine tool.

This property of automatic machine tools having cyclic operative mechanisms (cam-controlled) is manifested with especial expedience in machining small workpieces, in which case the particular cycles are of short duration. Since the lengths of the strokes are short as well as their duration, the savings in time due to clear-cut limiting of the travel are of a larger relative value than for longer strokes.

Positive continuous kinematic cyclic interlinkage between the working members of the machine enables the cycles of the various working members to be co-ordinated and their motions to be correlated with a precision that is difficult to attain when the operative mechanisms are of the noncyclic type.

Since all the particular cycles of the main and auxiliary working members are executed due to the cyclic changes in the kinematic linkages between the camshaft and the working members, and since the cyclic interlinkages between the working members are of a positive kinematic nature, the functions of controlling the operative mechanisms of the working members to accom-

plish the automatic cycle are reduced to naught for systems designed according to the given block diagram. The control functions may be extended to cover transmitting mechanisms T for the drive of the camshaft and spindle.

Since the general automatic cycle is co-ordinated in hundredths or in degrees of rotation of the camshaft, the control system operates with time-sequence control expressed in fractions of a camshaft revolution. If there is a positive continuous linkage between the camshaft and the working members, this is equivalent to a system with in-travel control of the working members.

The block diagram (Fig. 12) has been drawn up and the general properties of automatic machine tools with cyclic operative mechanisms have been considered here on the basis of structural properties common to all kinds of cyclic operative mechanisms. The structure of automatic machine tools with cyclic operative mechanisms (cam-controlled) can be treated more specifically only if the features of cyclic operative mechanisms of various kinds, and primarily cam mechanisms, are taken into account.

2-4. Cam Mechanisms

Figure 13 is a diagram of an elementary cam mechanism. The axis of cam rotation is offset from the line of action of the follower. The following notation has been accepted in this diagram (and subsequently):

Θ = pressure angle—the angle between the normal to the cam curve (profile) at the point of contact and the line of action of the follower

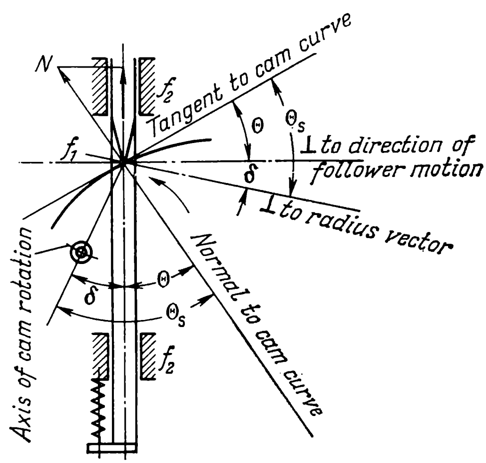


Fig. 13. Diagram of an elementary cam mechanism

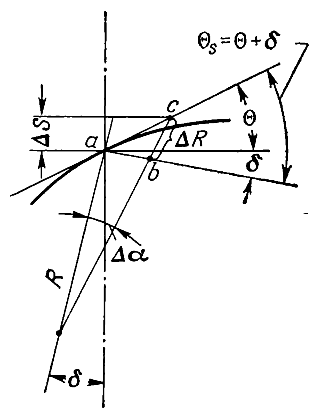


Fig. 14. Dependence of the central angle of cam rotation on the pressure angle

δ = offset angle—the angle between the radius vector and the line of action of the follower
Θ_s = slope angle of the cam curve—the angle between the radius vector and the normal to the cam curve (or between the perpendicular to the radius vector and the tangent to the cam curve at the point of contact)
P = force acting in the direction of follower motion
N = force normal to the cam curve.

The force growth factor $\varepsilon = \frac{N}{P}$ serves as the force characteristic of a cam mechanism. The normal force N, the bearing stress on the working surface of the cam, and cam wear increase with ε. Factor ε increases with the pressure angle Θ and slope angle Θ_s of the cam curve. As a rule, $\varepsilon = 1.35$ to 2.

Upon an increase in angle Θ_s, there is a danger of the cam mechanism being jammed as a result of the increased friction forces in the follower guides and on the cam surface, reducing the efficiency η_c of the cam mechanism to $\eta_c = 0$.

In the case of an elementary cam mechanism with no offset between the axis of cam rotation and the line of follower action, i.e., when $\delta = 0$ and $\Theta_s = \Theta$, jamming begins at the value

$$\Theta_{jam} = \text{arc tan}\,\frac{1 - f_1 f_2}{f_1 + f_2} \tag{1}$$

where f_1 and f_2 = coefficients of friction on the cam surface and in the follower guides, respectively
Θ_{jam} = pressure angle at which jamming occurs.

The safety factor of such a cam mechanism in respect to jamming can be written as

$$k = \frac{\tan \Theta_{jam}}{\tan \Theta} = \frac{1 - f_1 f_2}{(f_1 + f_2)\tan \Theta} \tag{2}$$

Thus a reduction in the pressure angle Θ leads to a reduction in the normal force N, acting on the working surface of the cam, and in the danger of jamming of the mechanism.

On the other hand, a reduction in angle Θ leads to an increase in the central angle of cam rotation required to obtain the given length of follower motion for the given overall size of the cam.

As can be seen in Fig. 14, if the cam rotates through a small angle $\Delta\alpha$, with sufficient accuracy for practical purposes

$$\Delta R \cong ab \tan(\Theta + \delta) \cong R\Delta\alpha \tan(\Theta + \delta)$$

the follower displacement being

$$\Delta S = \Delta R \cos \delta = R\Delta\alpha \tan(\Theta + \delta) \cos \delta$$

from which

$$\Delta\alpha = \frac{\Delta S}{R \tan(\Theta + \delta) \cos \delta} \tag{3}$$

With a reduction in the pressure angle Θ, the central angle of rotation $\Delta\alpha$ increases. At a definite speed of rotation of the cam, an increase in angle $\Delta\alpha$ will require more time for the given displacement of the follower, i.e., it will lead to a reduction in the production capacity of the automatic machine tool. This is of especial importance for rapid idle motions of the main working members (for example, slide approach and withdrawal). It is also taken into consideration when auxiliary working members have a cam drive.

The magnitude of the central angle of rotation of the cam serves as the principal index of the time required for the working and idle motions of the main and auxiliary working members of the machine. This should be taken into account in assigning the pressure and slope angles of the cam curve.

It can be seen from equation (3) that

$$R = \frac{\Delta S}{\Delta\alpha \tan(\Theta + \delta) \cos \delta} \tag{4}$$

Consequently, an increase in the pressure angle enables the overall size of the cam mechanism to be reduced.

In common practice, $\Theta = 10°$ to $40°$ for the curves of work cams while $\Theta_{max} = 45°$ to $60°$ for the curves of idle travel or return cams. These data concern the working and idle sections when their curves are on a single cam.

When the cam mechanism being emloyed differs from the elementary cam mechanism, the following equivalent coefficients of friction are introduced into equations (1) and (2)

$$f_{eq1} = \lambda_1 f_1 \text{ and } f_{eq2} = \lambda_2 f_2$$

where the reduction factors λ_1 and λ_2 are determined by comparison with the elementary cam mechanism.

To increase the safety factor in respect to jamming, and to increase the pressure angle with a consequent reduction in the central angle of cam rotation, measures should be taken to reduce the friction of the follower shoes and in the follower guides.

Instead of sliding shoes (flat-face followers), follower rolls (Fig. 15) are used, in which case

$$f_{eq1} = \lambda_1 f_1 = \frac{d}{D} f_1$$

or rolls mounted on rollers, needle bearings or antifriction bearings (Fig. 16).

The value of the central angle of rotation can be varied by offsetting the axis of rotation of the cam, since an increase in the offset angle δ (Fig. 13) reduces the central angle of rotation $\Delta\alpha$ [see equation (3)].

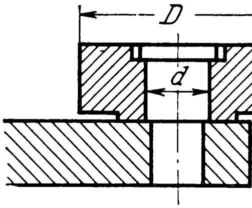


Fig. 15. Follower roll with a plain bearing

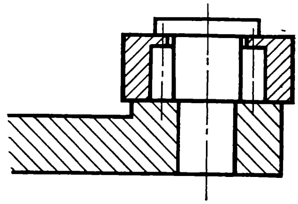


Fig. 16. Follower roll with an anti-friction bearing

In practice, the maximum overall size of the cams has been established for each existing model of machine tool and is dictated by the maximum diameters of the cam blanks. In designing setups for machine tools under these conditions, it is good practice to use curves with the maximum possible radii for the working travel. This leads to a reduction in the pressure angle Θ and in the normal force growth factor ε on the cam surface.

Adjusting the length of follower travel. To change over a cam-controlled automatic machine tool for machining workpieces of different sizes, it is usually necessary to change the lengths of travel of the main and sometimes the auxiliary working members.

With the general trends being to extend the field of application of an automatic machine tool and to simplify and speed up changeovers, the choice of a method for adjusting the stroke depends, not so much on the type and volume of production, as on the conditions under which cam mechanisms are being employed for driving the various working members.

Interchangeable Cams

Cams are changed to set up an automatic machine tool for machining each different workpiece, or for each of two, three or several size groups of workpieces extending over the whole field of application of the machine.

In the first case, the interchangeable cam effects complex movements of the main working member (tool slide), required for form machining (single-edge cylinder feed cam for the headstock of a Swiss-type automatic), or it accomplishes a number of consecutive operations (or operation elements) performed by a single slide actuated by a single cam (cam of the turret slide in an automatic screw machine or of the tool slides in a Swiss-type automatic). In this case the cam is of strictly individual design.

In the second case, the operating cycle is simple and, in addition to the interchangeable cams for setting up to the size group of workpieces, other devices are available for adjusting the stroke in setting up the machine for a definite workpiece within the size group. Interchangeable cams of this

type are employed, for instance, for actuating the end tool slide of multiple-spindle automatics.

Variable-lift cams (Fig. 17) are rarely employed because the law of motion of the follower is violated when the cam is adjusted, slide travel becomes irregular, and the construction of the cam is insufficiently rigid. Such cams are difficult to adjust.

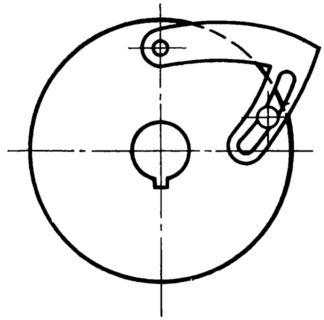


Fig. 17. Diagram of a variable-lift cam

Utilizing only a part of the working travel. The tool may be set up so that cutting occurs only during part of the working travel; during the remainder, the tool is advanced to the work by the working feed curve of the cam. This is poor practice, in general, and can be justified only if the tool approach coincides with the working travel of other tools or if it takes place during rapid rotation of the cam, as in the case of group setups of automatic machine tools (see Sec. 8-6).

In the drives of auxiliary working members, at high speeds of traverse, partial utilization of the working travel leads to lower losses of time. An example of this is the stock feed to a stop in certain multiple-spindle automatics.

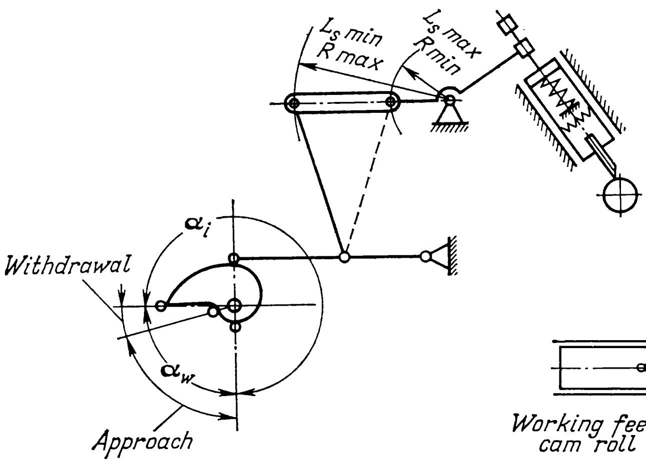


Fig. 18. Adjusting the slide stroke by varying the arm length of the driven lever

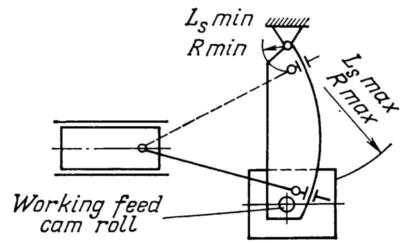


Fig. 19. Adjusting the slide stroke by varying the arm length of the driving lever

Adjusting the stroke length by varying the arms of levers transmitting motion from the cam to the slide. To increase the length of slide travel, or stroke, the tie-rod is fastened at the smaller radius of a driven lever (Fig. 18) or at the larger radius of a driving lever (Fig. 19). This system of stroke adjustment has the following drawbacks: the length of slide approach is varied together with that of the stroke; the rate of approach and withdrawal and the working feed are varied together with the stroke, though the times for the working and idle travel remain constant; the nonuniform feed inherent in a lever system with a crank and tie-rod makes it necessary to use spring compensators in the feed drive of thread-cutting die heads.

The advantages of this system are the considerable range of stroke length adjustment available, simple and convenient adjustments and the dependability of the adjusting devices.

This method is widely applied for adjusting the stroke length of the cross slides of automatic screw machines and multiple-spindle automatics, since the length of approach of these slides is not very great and its variation is of no practical significance, especially when the approach and withdrawal coincide in time with other operations or elements.

Duplex Cams

If the working stroke of a slide alternates with an indexing cycle to change the position of the workpiece (spindle carrier of multiple-spindle automatics) or of the tools (turret of automatic screw machines), then the slides with tools arranged coaxially with the workpiece must be rapidly withdrawn from the machining zone, to avoid collisions in indexing, and then rapidly advanced again into the machining zone.

Special devices are used in such cases for rapid approach to and withdrawal from the cutting zone. An additional rapid approach and withdrawal cam, paired with the working feed cam, is used for the end tool slide of multiple-spindle automatics; a special crank gear mechanism, powered by a high-speed drive, is used to advance and retract the turret slide.

Duplex (positive-return) cams are mounted on the camshaft of an automatic adjacent to each other. The length of working travel is adjusted by varying the arms of levers that transmit motion from the working feed cam to the slide. The length and rate of approach actuated by the slide approach cam remain constant. To increase the rate of approach and withdrawal, the camshaft, carrying the duplex cams, rotates at a higher speed, as a rule, than it does during working feed of the slide actuated by the working feed cam.

Duplex cams may be of either the drum (cylinder) or disk (plate) type.

Figure 20 illustrates schematically the drive of end (main) tool slide *2* in the model 1240-6 six-spindle automatic accomplished by means of working feed cam *6* and rapid approach cam *8*, mounted adjacently on the cam-

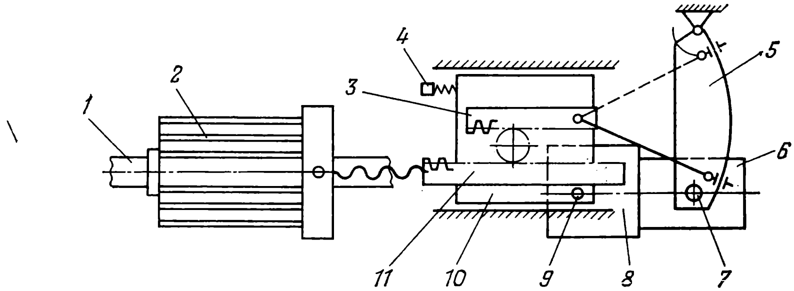


Fig. 20. End tool slide drive mechanism with duplex cams:
1—carrier stem; *2*—end tool slide; *3*—tie-rod rack; *4*—stop screw; *5*—lever; *6* and *7*—working travel cam and roll; *8* and *9*—rapid approach cam and roll; *10*—carriage; *11*—rack; *12*—hand adjustment screw

shaft. The length of travel at the working feed is set up by adjusting the point at which the end of the tie-rod is fastened to lever *5*. This mechanism operates on the following principle.

Approach of tool slide 2. Rapid rotation of the camshaft carrying the duplex cams.

Lever *5* and tie-rod rack *3* are stationary since roll *7* of lever *5* runs along a dwell curve of cam *6*. Carriage *10* travels to the left, being actuated by approach cam *8* along whose curve runs roll *9* of the carriage. Travelling together with carriage *10* is the pin of the rack pinion which rolls along stationary rack *3* and advances rack *11*, linked to tool slide *2*, at twice the speed of carriage *10*.

Working feed. Slow rotation of the camshaft.

Carriage *10* is held against stop *4* since its roll *9* runs along a dwell curve on cam *8*. Lever *5* is swung by cam *6* to the right, shifting rack *3* to the right. This motion is transmitted through the pinion to rack *11* and tool slide *2* which travel to the left at the rate of the working feed. Screw *12* sets up the extreme left-hand position of the tool slide.

Tool slide withdrawal. Rapid rotation of the camshaft.

Lever *5* and rack *3* are shifted to the left by cam *6*. Carriage *10* and the rack pinion pin are traversed rapidly to the right by cam *8*. Rack *11* and tool slide *2* travel rapidly to the right at the summed speed.

Drive of the Independent-Feed Tool Spindles in the Model 1240-6 Automatic

The travelling holders of the high-speed drilling and threading spindles travel along slots of the end tool slide of this six-spindle automatic (Fig. 21) at a rate of feed independent of that of the tool slide.

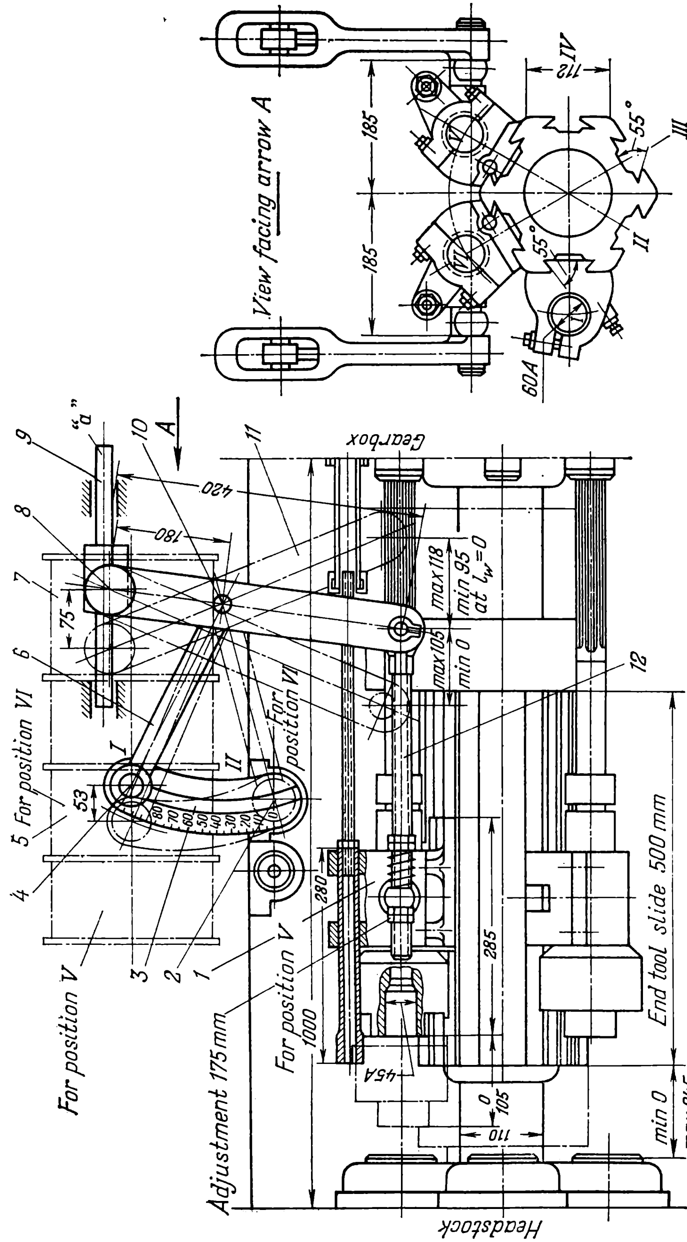


Fig. 21. Drive of the independent-feed tool spindles in the model 1240-6 automatic

Duplex cams with a lever transmitting mechanism are used for rapid approach and withdrawal of the travelling holders.

Holder *1* is actuated for the working feed motion by permanent cams mounted on drum *5* (at the left for position *V* and at the right for position *VI* of the spindle carrier); it is actuated for the approach and withdrawal motions by rapid traverse cams mounted on drum *7*. Roll *4* of sector *3* swings the sector to the left about axis *2*. This motion, through link *6*, lever *11* and adjustable tie-rod *12*, traverses the spindle holder to the left. At the same time, the roll of bar *9*, carrying pin *8* on which the end of lever *11* is mounted, is held stationary by the cams of approach drum *7*.

In the approach motion, the roll of movable bar *9* shifts the bar and pin *8* of lever *11* to the right. At this, roll *4* of sector *3* is held stationary by the cams on working feed drum *5*. In spindle holder withdrawal, sector *3* swings to the right, while pin *8* of lever *11* is shifted to the left together with bar *9*.

The length of travel at the working feed and the rate of feed are reduced by adjusting the pin of link *6* along the slot of the sector toward axis *2* (about which the sector swings).

Duplex cams are frequently of the disk (plate) type. In the model 1A225-6 six-spindle automatic, a section of the camshaft is arranged square to the axis of the machine and the end tool slide is actuated by duplex disk cams (Fig. 22).

The working feed of tool slide *1* is effected by the working feed disk cam through lever *7*, pivoted on pin *8*, link *10*, rocker arm *3* and adjustable tie-rod *2*. The length of working travel is set up by adjusting pin *11* of link *10* along arcuate slot *9*.

In the approach motion, the approach cam turns lever *6* about pin *5*. This shifts pin *4* of rocker arm *3* to the left, advancing the tool slide into the cutting zone; at the same time, lever *7* is held stationary.

At the end of the approach motion, lever *6* runs against stop *12* and is rigidly fixed between the stop and the cam. It remains in this fixed position throughout the working travel.

In the withdrawal motion the rolls of levers *6* and *7* are swung outward (away from the axis of the camshaft). Pin *4* of rocker arm *3* is shifted to the right as is the lower pin of rocker arm *3*.

Duplex cams have the following advantages over a slide travel drive with a single cam in which the length of working travel is set up by adjusting the arms of levers in the lever transmitting system:

(1) the length and rate of approach remain constant regardless of adjustments in the length of working travel;

(2) a large total length of slide travel can be obtained with a sufficiently large approach and withdrawal.

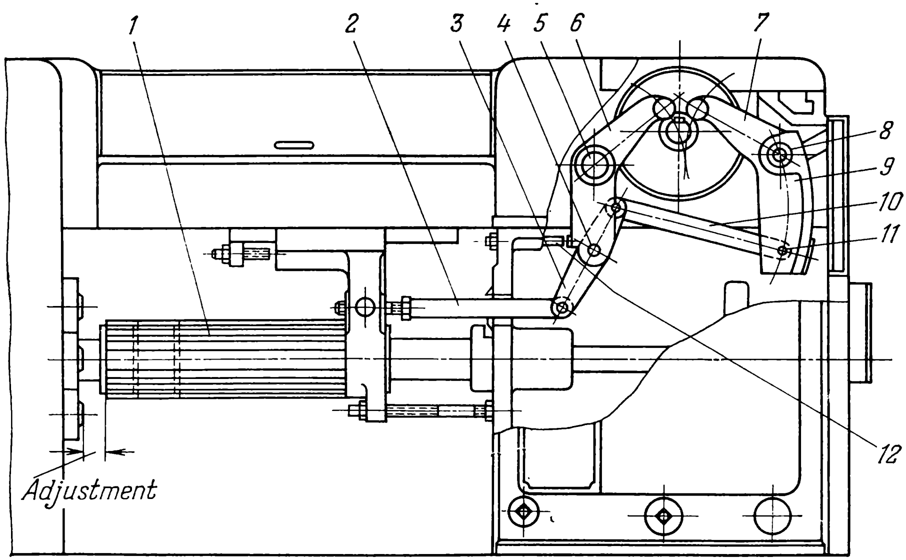


Fig. 22. Drive of the end tool slide in the model 1A225-6 automatic

The main field of application of duplex cams is in the feed drives for the end tool slides of multiple-spindle automatics and for the independent-feed tool spindles mounted on these tool slides.

During the approach and withdrawal motions, the camshaft carrying the duplex cams should rotate at high speed.

2-5. Structural Properties of Cam Mechanisms

In addition to the structural properties common to all cyclic mechanisms (see Sec. 2-2), cam mechanisms are distinguished by certain features that substantially affect the structure of the automatic machine tool.

1. The effect of the large "pitch", i.e., feed per revolution of the camshaft, on the structure of the drive mechanism.

The "pitch" of the cam is

$$t = \frac{360^\circ l_s}{\alpha_w^\circ i_c} \tag{5}$$

where l_s = length, mm, of slide travel at the working feed

α_w = central angle of rotation of the cam, deg, during the working travel of the slide

i_c = transmission ratio from the cam to the slide.

However, since $\alpha_w^\circ < 360°$,

$$t > l_s$$

It follows from the kinematic balance equation

$$1s = i_{sp/s} t \tag{6}$$

that

$$i_{sp/s} = \frac{s}{t} = \frac{s\alpha_w^\circ i_c}{360° l_s} \tag{7}$$

where $i_{sp/s}$ = ratio of the gear train from the spindle to the traversing shaft which coincides with the camshaft

s = feed, mm per spindle revolution.

Since the value of t is large, the degree of reduction in the feed train from the spindle to the traversing shaft (camshaft) is always very large, and one and, more frequently, two sets of worm gearing are required in the train, of which one set is adjacent to the camshaft, so as to reduce the torque on the preceding shafts.

The torque on the traversing shaft [see equation (104) Vol. 3] is

$$M_t = \frac{Qt}{2\pi\eta_s} \text{ kgf-mm} \tag{8}$$

where Q = feeding force, kgf

t = "pitch" of the traversing device, mm

η_s = efficiency of the traversing device.

It follows from equation (8) that the torque on the traversing shaft (camshaft) of a cam mechanism is very large due to the large value of t, the "pitch" of the traversing device.

2. It was mentioned before that, as a cyclic operative mechanism, a cam mechanism is distinguished by the fact that the law of motion of the working member can be assigned arbitrarily, within certain limits, by suitably designing the cam curve. This means that engagement and disengagement of the travel of the working member, its path length, velocity and law of velocity variation, direction of motion, changes in direction and rest periods (dwell) can be varied at will by providing cam curves of the corresponding shapes. This imparts adaptability to a cam mechanism in co-ordinating the times of engagement and disengagement of the motions of the various working members in the general automatic cycle (see Sec. 2-2). Within the limits of the particular cycle of a single working member, each elementary motion of the cycle can be accomplished according to the optimum law by profiling the cam in the corresponding manner.

For these reasons, the cam mechanism is the principal cyclic operative mechanism used for the main and auxiliary working members of automatic machine tools.

3. The structure of cam-controlled automatic machine tools depends upon the method used to maintain contact between the cam and the follower in the operative mechanisms of the main working members.

Two different systems of maintaining contact are distinguished:

Spring-action contacting, in which the roll is held against the cam by a spring (rarely by a weight) located in the carriage and acting against the slide carrying the tools.

Kinematic contacting, in which the roll is held against the cam by the reaction of the cutting forces during the working travel and by the forces of inertia during idle travel.

In spring-action contacting, the tool slide is traversed for withdrawal by spring action (Fig. 23). The roll runs along a fall curve which governs its law of motion but does not traverse the slide. Hence, the fall, or drop, curve is arranged at a slope angle near to 90°. The slide is withdrawn during a very small central angle of cam rotation (usually 2 to 6 hundredths of a revolution). Thus, in spring-action contacting, the slide is withdrawn sufficiently rapidly at a constant speed of rotation of the cam throughout the cycle, no high-speed rotation of the camshaft during the withdrawal motion being required.

Spring-action contacting can be applied for a limited length of withdrawal (maximum of 120 to 200 mm), since springs do not operate efficiently at longer travel.

If the tool slide is very heavy and the forces of inertia are great, spring operation involves heavy impacts at the end of the withdrawal motion and, for this reason, spring-action contacting is not applicable in such cases. It is suitable for small machine tools using plate cams with a rise not exceeding 200 mm.

As mentioned above, rapid (high-speed) rotation of the camshaft is not needed for slide withdrawal when spring-action contacting is used. If, in this case, the length of approach is small (angle γ is small in Fig. 23), the whole machining cycle can be carried out at constant speed of camshaft rotation.

In kinematic contacting (Fig. 24), the slide is withdrawn by a return cam with a curve slope angle not exceeding 55°, the angle of rotation of the camshaft being considerable. Therefore, to obtain a sufficiently high speed of slide withdrawal, the cam, and consequently the camshaft, should rotate at a speed higher than that during the working feed. This complicates the control system and the drive of the camshaft in the automatic machine tool.

Kinematic contacting is used in medium-sized machine tools with slide travel exceeding 200 mm, when the slides are quite heavy, i.e., in cases when spring-action contacting is inapplicable. It is usually employed with drum cams which provide slide travel lengths up to 300 mm.

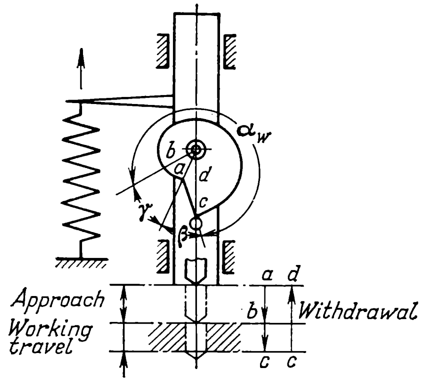


Fig. 23. Spring-action contacting in a cam mechanism:

γ—angle of approach; α_w — angle of working travel; β—angle of withdrawal

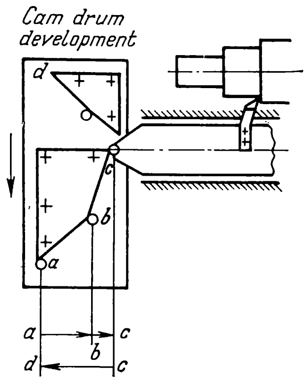


Fig. 24. Kinematic contacting in a cam mechanism:

a-*b*—slide approach; *b*-*c*—working travel of slide; *c*-*d*—slide withdrawal

4. The length of slide approach in the cutting zone is of great structural significance. Both in spring-action contacting (angle γ in Fig. 23) and in kinematic contacting (cam curve *ab* in Fig. 24) the slide is advanced by the action of the cam with a limited slope angle of the curve. Nevertheless, if spring-action contacting is used and the length of approach is not too large, it is possible to do without high-speed rotation of the camshaft. Examples are the Swiss-type automatic screw machines and automatic cutting-off machines, and the cross slides of automatic screw machines.

If the working travel of the slide alternates with an indexing cycle to change the position of the workpiece (spindle carrier of a multiple-spindle automatic), then duplex cams are resorted to for approach and withdrawal of slides carrying tools coaxial with the workpiece (see page 34 and Figs. 20, 21 and 22).

The use of duplex cams leads to an increase in the angle of camshaft rotation for the working cycle and requires high-speed rotation of the camshaft during idle travel motions of the end tool slide.

In an automatic screw machine, a special crank gear mechanism driven from a high-speed auxiliary shaft is used for rapid approach and withdrawal of the turret slide (see Fig. 42). The function of the plate cam of the turret slide is limited to travel at the working feed and rapid withdrawal with spring-action contacting along the fall curve of the cam to the position at which the next working travel motion begins. As a result, the plate cam may provide six consecutive cycles of working and idle travel movements

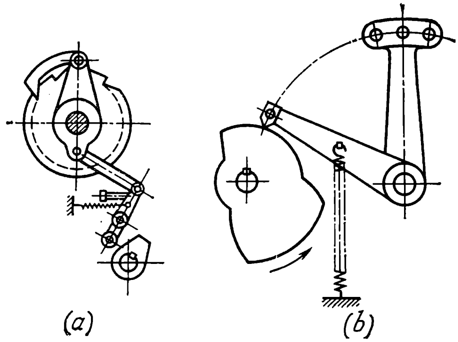


Fig. 25. Indexing mechanisms: (a) cam-ratchet wheel; (b) cam-lever

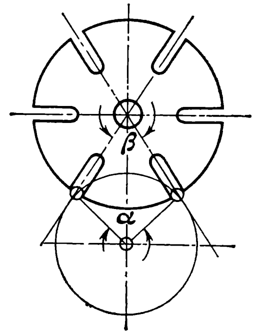


Fig. 26. Diagram of a Geneva wheel mechanism

of the turret slide during the general cycle of the automatic, while the need for high-speed rotation of the camshaft is excluded.

As elements of indexing mechanisms, cams are used in conjunction with ratchet wheel mechanisms and with levers (Fig. 25*a* and *b*). They operate with a small central angle of cam rotation for an indexing motion and do not require high-speed rotation of the camshaft.

The cam curves can be designed so as to change the law of motion in the indexing cycle in the desired manner.

2-6. Structural Properties of Other Cyclic Operative Mechanisms

Geneva wheel mechanism. Under the condition that the roller or pin enters (Fig. 26) and leaves the wheel slot without impact, the central angle α of driver rotation during one indexing motion (wheel rotation through the angle β between adjacent slots) should be supplementary to the angle between slots, i.e.,

$$\alpha + \beta = 180^\circ$$

(see page 166, Part Five, Vol. 3). The angle of driver rotation $\alpha = 90^\circ$ for a four-slot Geneva wheel and $\alpha = 120^\circ$ for a six-slot wheel.

These large values of the central angle of rotation make it necessary to power the driver by a high-speed auxiliary shaft (as in the case of the turret of an automatic screw machine, Fig. 42) or by a separate drive with an individual electric motor (spindle carrier of semiautomatic multiple-spindle vertical chucking machines). If the driver of the Geneva wheel is mounted on the camshaft, high-speed rotation is transmitted to this shaft during wheel

indexing (spindle carrier indexing in horizontal multiple-spindle automatics).

As a measure for reducing the angle of driver rotation, a four-slot Geneva wheel is used for the six-spindle carrier as well, the driver being arranged on the camshaft and gearing with a ratio $i = {}^2/_3$ being provided in the transmission to the carrier and to the stock reel.

Linkgear mechanism. Linkgear mechanisms are used, in the same manner as Geneva wheels, to provide the indexing cycle of the carrier of vertical multiple-spindle semiautomatics. The angle of crank rotation in the indexing cycle is at least 120°, so that the crank is usually driven by a high-speed auxiliary shaft.

Crankgear mechanism. Such mechanisms operate with a large angle of crank rotation (180°) per stroke and therefore should be driven by a high-speed auxiliary shaft.

2-7. Structural and Kinematic Diagrams of Cam-Controlled Automatic Machine Tools

The structure of a machine tool is dictated by the kind of workpiece and its machining techniques; these factors determine the nature of the cycles of the working members and are the basis for selecting the operative mechanisms that can accomplish these cycles.

A structural block diagram is developed to reveal the structure of a definite automatic machine tool, and to take into consideration the structural features of the cam and other cyclic operative mechanisms, the effect of these features on the transmission (drive) mechanism of the machine tool, and the functions of the automatic cycle control system.

The development of such a block diagram is preceded by the selection of the operative mechanisms for the working members of the machine. In their turn, the operative mechanisms affect the structure of the camshaft drive mechanism.

The structural block diagram serves as the basis for working out the kinematic scheme of the automatic machine tool.

The structures of cam-controlled automatic machine tools can be classified into three groups.

The *first structural group* is characterized by the use of disk (rarely drum) cams with spring-action contacting as operative mechanisms of the main and auxiliary members. As a result, the camshaft of the automatic machine tool rotates during the whole cycle at a constant speed set up in accordance with the cycle time.

Automatic machine tools of this group are designed for machining small-size workpieces not requiring long strokes of the slides.

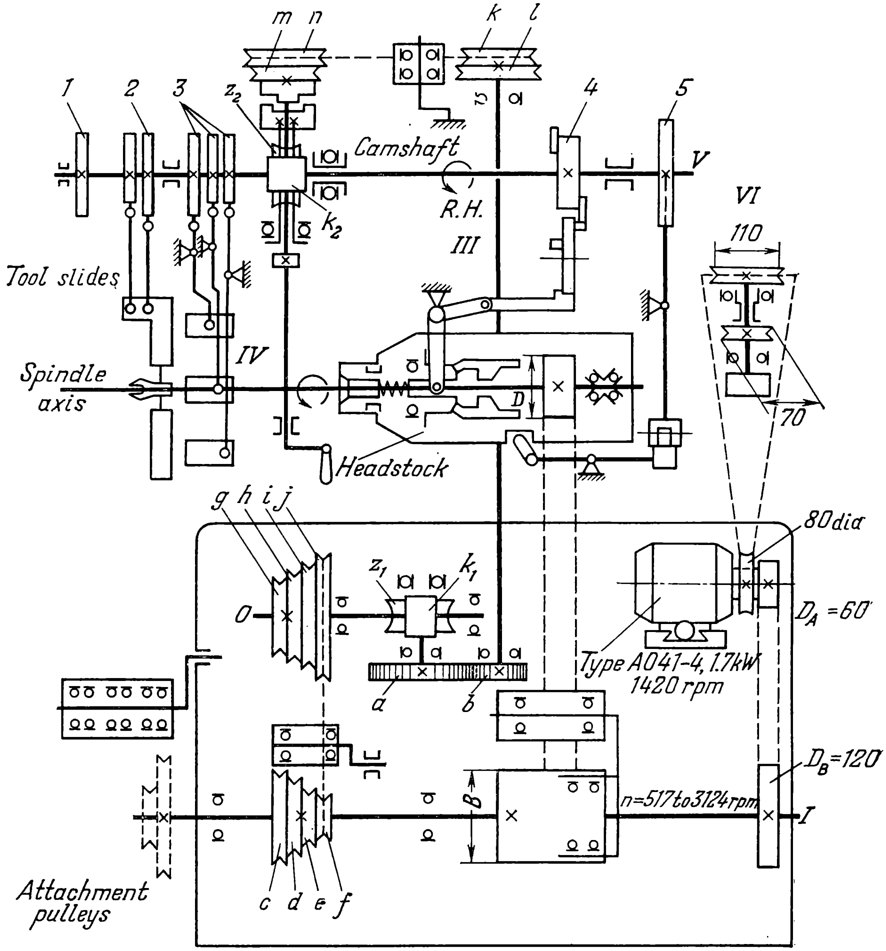


Fig. 27. Kinematic diagram of the model 1A10П automatic:
1—cam for indexing the drilling attachment (see Fig. 25*b*); *2*—cams for actuating the rocker arm tool slides (see Fig. 29); *3*—cams for actuating the vertical tool slides (see Fig. 28); *4*—chucking (collet operating) cam; *5*—plate cam for headstock feed (see Fig. 31, item *2*)

Automatics of this group are of the single-spindle type, as a rule, without turrets. They do not have rotary indexing cycles for changing positions that require the application of a Geneva wheel mechanism. The swivelling indexing cycle, required to change the spindle positions of the high-speed

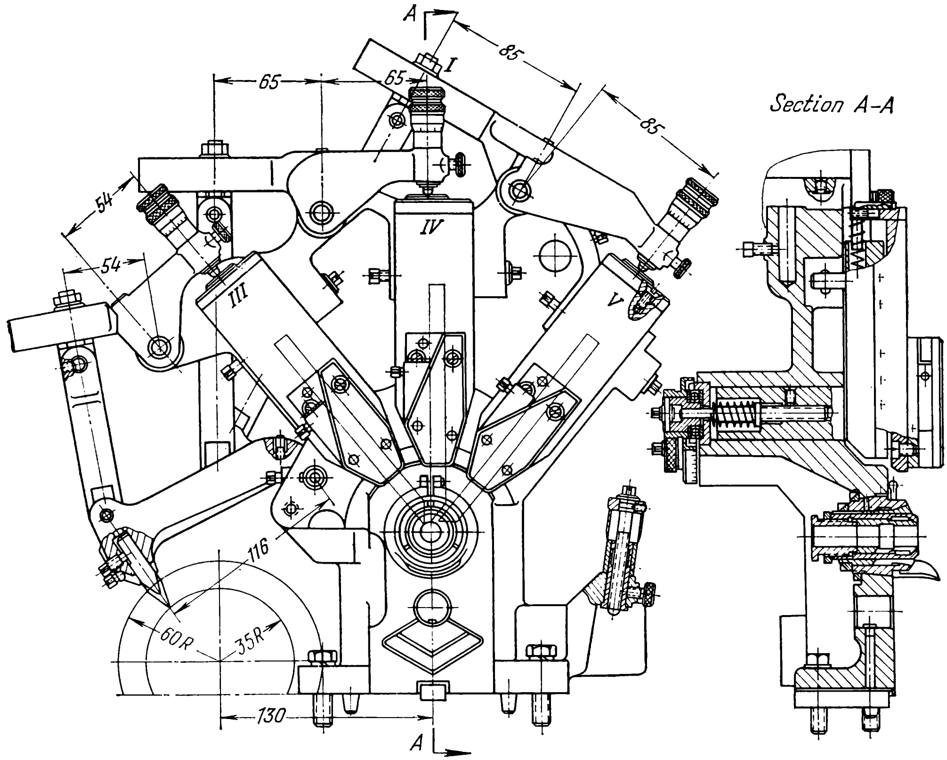


Fig. 28. Vertical tool slides of the model 1A10П automatic

drilling and thread-cutting attachments, is effected by a cam mechanism during a small central angle of cam rotation (Fig. 25*a* and *b*).

Workpieces may be of quite intricate configuration, requiring a complex cycle of tool slide motions consisting, in such cases, of several repeated approaches, working strokes, dwells and withdrawals.

Large lengths of slide approach and withdrawal are not necessary in automatics of this group (see pages 34 and 40). This excludes the need for duplex cams and, consequently, for high-speed rotation of the camshaft during the approach and withdrawal motions.

The kinematic diagram of the model 1A10П precision Swiss-type automatic screw machine is shown in Fig. 27 as an example of the first group of automatic machine tools.

This automatic turns work of a diameter up to 7 mm with a diametral accuracy of the 1st grade and a length accuracy of the 2nd grade according to the USSR standard.

The automatic turns work of cold-drawn and sized steel bar stock, preferably after it has been ground in a centreless grinder to a diametral tolerance within 0.01 mm.

The high accuracy attainable in Swiss-type automatics is due, in the main, to the machining method applied. The stock is fed together with the headstock (or the spindle quill) and is turned adjacent to the tool frame in a rotating guide bushing by single-point tools held with small overhang in the cross tool slides (Figs. 28 and 29). The headstock is retracted by a spring for clamping the stock. During this retraction, cut-off tool withdrawal is delayed after cutting off the finished part, and the bar stock is held against the cut-off tool by a pusher which is actuated by a counterweight through pulleys and a wire rope.

Headstock *1* with the bar stock is fed forward for working travel by single-edge cylinder cam *2* directly, with no transmission members, and without

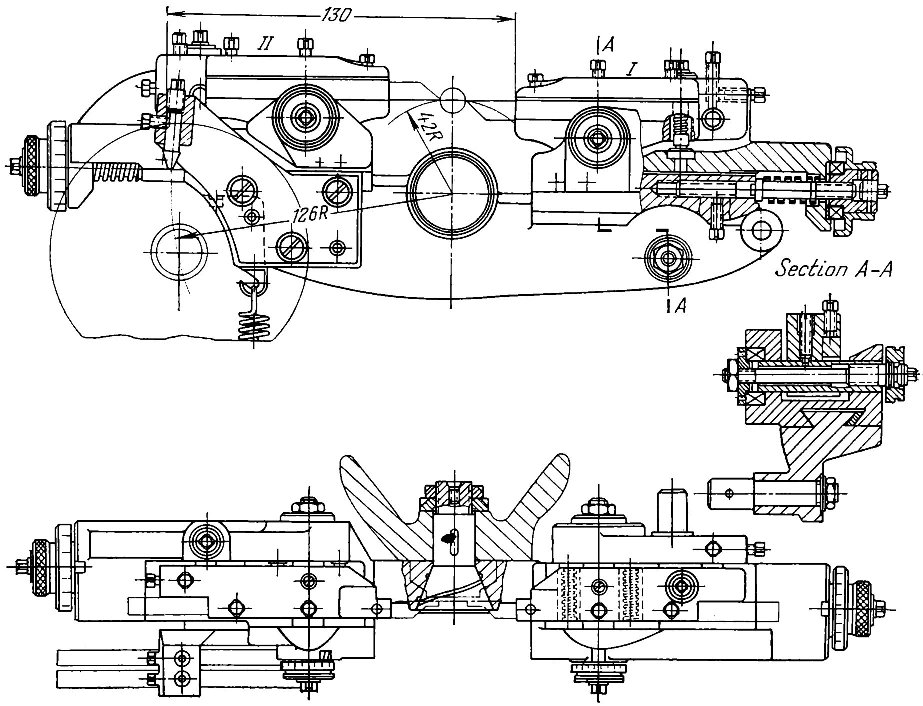


Fig. 29. Rocker arm slides of the model 1A10П automatic

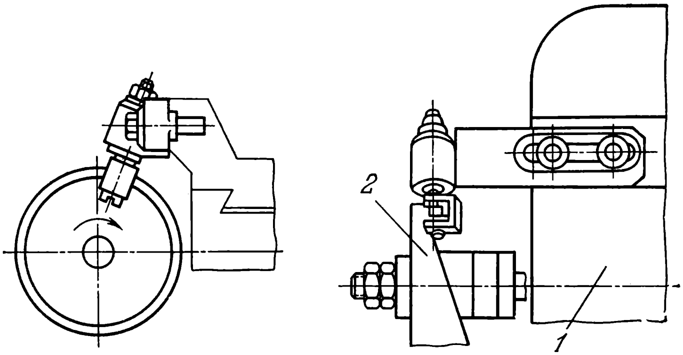


Fig. 30. Mechanism for traversing the headstock with a single-edge cylinder cam

any violation of uniformity or any other law of the feed motion provided by the cam curve (Fig. 30). This is of significance in form turning by a combination of longitudinal and cross feeds.

In addition to the single-edge cylinder cam, the same model has disk cam *2* (Fig. 31) for traversing the headstock. Motion is transmitted through a lever system with facilities for varying the length of headstock travel by changing the arm length of lever *3* linked to the headstock. The rate of feed, in this case, varies along the length of travel but it is possible to change over the automatic for turning workpieces of like shape, but of different lengths, without changing the cams (for example, for turning screws of different lengths).

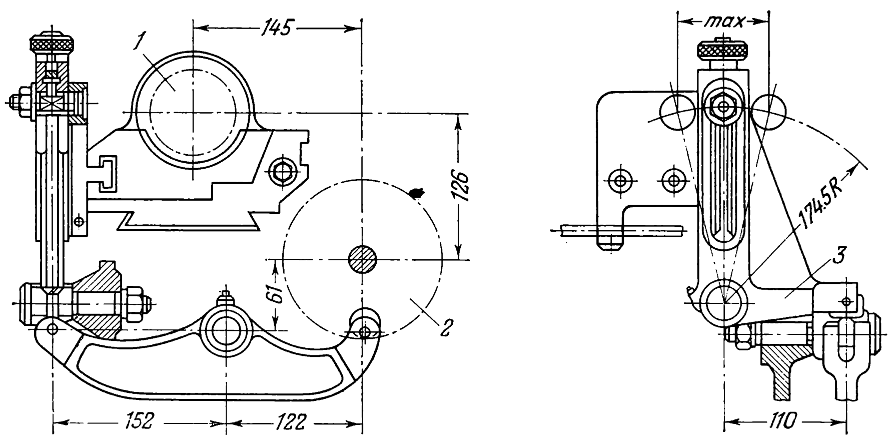


Fig. 31. Mechanism for traversing the headstock with a disk cam

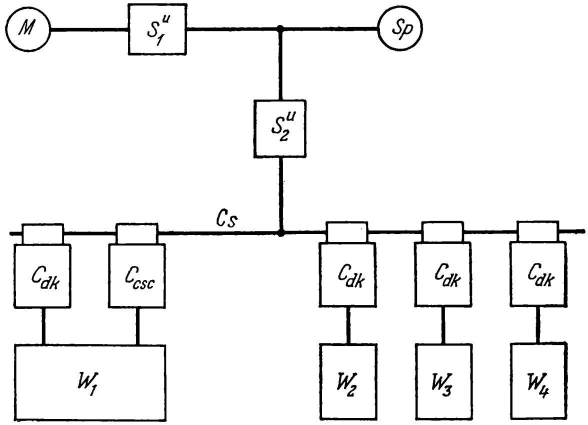


Fig. 32. Structural block diagram of the model 1A10П automatic

The structural block diagram of the automatic is illustrated in Fig. 32. The notation is as follows: M—electric motor of the main drive; Sp—spindle; S_1^u—device for setting up the spindle speeds; S_2^u—device for setting up the camshaft speed; Cs—camshaft; C_{dk}—disk (plate) cam; C_{csc}—single-edge cylinder cam with spring-action contacting; W_1—headstock; W_2—rocker with two tool rests; W_3—three vertical tool rests and W_4—stock clamping device.

The cam operative mechanisms have the spring-action contacting feature and do not require high-speed rotation of the camshaft. The disk cam of the clamping mechanism operates with a small central angle of cam rotation. This can be seen in the structure of the camshaft drive in which there is only one drive train with the setting-up facilities S_2^u used to obtain the required camshaft speed so that the camshaft makes one revolution during the machining cycle for one workpiece. Calculations for setting up the camshaft speed are based on the kinematic balance equation (see Fig. 27)

$$1 \text{ rev } (Cs) = \frac{z_2}{k_2}\frac{n}{k}\frac{b}{a}\frac{g}{c}\frac{z_1}{k_1}\frac{B}{E} = n_{sp}^{wp} \tag{9}$$

where

Cs = camshaft

$\frac{n}{k}$ and $\frac{m}{l}$ = available alternate belt drive ratios

$\frac{g}{c}, \frac{i}{e}, \frac{h}{d}$ and $\frac{j}{f}$ = available alternate belt drive ratios

n_{sp}^{wp} = number of spindle revolutions in machining one workpiece.

At the same time

$$n_{sp}^{wp} = n_{sp} T$$

where T = machining cycle time in min, calculated from the design setup operation chart on the basis of the sequence of manufacturing operations

n_{sp} = spindle speed in rpm.

It follows from the kinematic balance equation (9) that for the drive setup

$$\left(\frac{k}{n} \text{ or } \frac{l}{m}\right)\frac{a}{b}\left(\frac{c}{g} \text{ or } \frac{d}{h} \text{ or } \frac{f}{j} \text{ or } \frac{e}{i}\right) = \frac{z_2}{k_2}\frac{z_1}{k_1}\frac{B}{E}\frac{1}{n_{sp}^{wp}} \tag{10}$$

It is evident from the structural diagram (Fig. 32) that in automatic machine tools of this group the functions of the control system in obtaining the automatic cycle are reduced to naught (see page 28).

The *second structural group of cam-controlled automatic machine tools* is characterized by the use of drum cams (less frequently—plate cams) with kinematic contacting (see pages 40 and 42) and other operative mechanisms (Geneva wheel). These operative mechanisms require rotation of the camshaft at a constant high speed during the idle and auxiliary motions and slower rotation when the slides travel at the rate of working feed.

The speed of the camshaft is set up in accordance with the cycle time for machining one workpiece.

The second structural group is applied in semiautomatics for turning medium and large-size workpieces. Lengths of slide travel exceed 150 or 200 mm so that plate cams with spring-action contacting are inapplicable and drum cams with kinematic contacting must be resorted to.

For this reason, this structure is employed, for example, in single-spindle semiautomatic lathes for turning medium and large-size work and in semiautomatic fixed-bed milling machines with a table travel up to 300 mm.

High-speed rotation of the camshaft during the idle and auxiliary motions is applied in multiple-spindle automatics, as well as in horizontal semiautomatics.

With a cycle time $T_c = t_w + t_i$, where t_w is the time for travel at the rate of working feed and t_i is the time for idle and auxiliary motions, the share of time required for idle and auxiliary motion can be expressed as

$$\frac{t_i}{T_c} = \frac{t_i}{t_i + t_w} = \frac{1}{1 + \frac{t_w}{t_i}}$$

For the structure under consideration

$$t_i = \frac{1}{n_{cs}^{hs}} \frac{a_i}{360}$$

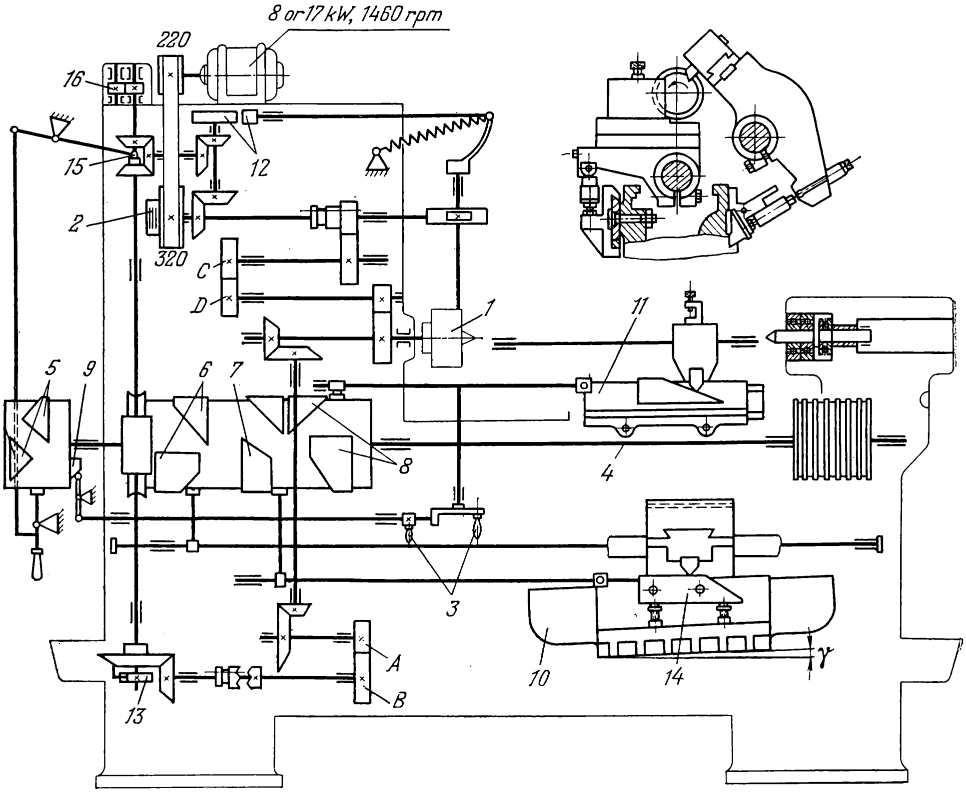


Fig. 33. Kinematic diagram of the model 116 semiautomatic lathe:

1—spindle; *2*—clutch for starting the lathe; *3*—control lever; *4*—camshaft; *5*—cams for engaging high-speed rotation of the camshaft; *6*—cams for longitudinal feed of the front tool slide; *7*—cams for the former slide of the front tool slide; *8*—cams for the former slide of the rear tool slide; *9*—cam for disengaging clutch *2*; *10*—former slide of front tool slide; *11*—former slide of rear tool slide; *12*—brake; *13*—overrunning clutch; *14*—former of front slide; *15*—high-speed clutch; *16*—pump

where n_{cs}^{hs} = speed of the camshaft during high-speed rotation, rpm
α_i = angle of rotation, deg, of the camshaft during the idle and auxiliary motions.

Since α_i and n_{cs}^{hs} are constant for a given machine tool with this type of structure, the share of time $\frac{t_i}{T_c}$ for idle and auxiliary motions is reduced with an increase in the cycle time. Hence, a machine tool with this structure can operate to greater advantage on longer cycles.

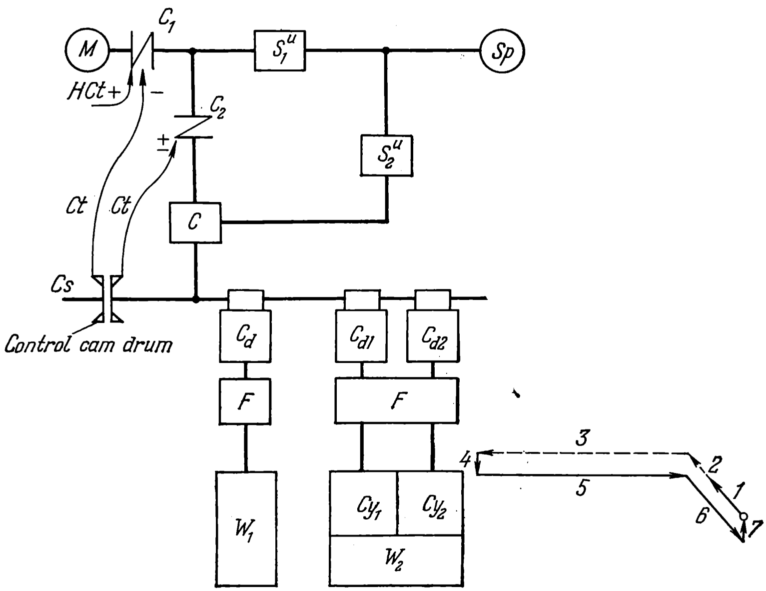


Fig. 34. Structural block diagram and operating cycle of the front tool slide in the model 116 semiautomatic lathe

Because of the comparatively large masses arranged on the camshaft, dynamic conditions limit the speed of idle camshaft rotation.

In machine tools of various sizes $n_{cs}^{hs} = 8$ to 30 rpm. In this case, the time required for the auxiliary motions may be larger than the minimum value permitted by the conditions of operation of the operative mechanisms of the auxiliary members. This is a disadvantage of the given structure.

The kinematic diagram of the model 116 single-spindle semiautomatic lathe is shown in Fig. 33. This lathe is intended for machining work of a diameter up to 300 mm. The corresponding structural block diagram is given in Fig. 34 where the following notation has been used:

C = conjunction of the trains for working feed and high-speed rotation of the camshaft by means of a nonreversible overrunning clutch

C_2 = engagement of the camshaft high-speed train by a friction clutch

Ct = function of control system

C_d = drum cam with kinematic contacting for actuating the former slide of the cross-feeding rear tool slides

F = travelling former

W_1 = rear (cross) tool slide
W_2 = front (longitudinal) slide
C_{d2} = drum cam for traversing the former slide along inclined guides to obtain part Cy_2 of the front tool slide cycle: tool relief (retraction) *4* and lowering of the former, dwell of the former, tool advance *7* with the former and dwell of the former
C_{d1} = drum cam for traversing the front tool slide to obtain part Cy_2 of the cycle: *1* —rapid inclined approach along the former; *2*—inclined feed-in along the former at the rate of longitudinal working feed; *3*—working feed as the shoe of the tool slide runs along the horizontal part of the former; *5*—rapid return along the horizontal part of the former; *6*—rapid inclined withdrawal along the inclined part of the former.

Upon horizontal travel of rear former slide *11* to the right (Fig. 33), the inclined part of the rear former actuates the shoe of the cross-feeding rear tool slide, swinging it forward in the bearings of the back bar, and thereby providing cross feed of the tool.

The front (longitudinal) tool slide, travelling during the working feed to the left together with the centre bar, bears with its shoe on the horizontal part of front former *14*. At the end of the longitudinal working feed, former slide cam *7* shifts the former slide to the left along inclined guides, as a result of which the horizontal part of the former and the slide shoe are lowered. At this, the front tool slide swings back with its bar and retracts the tool from the work (tool relief). After this, the former slide remains stationary while the front tool slide and its bar are returned rapidly to the right to the initial position. At the end of the return travel, inclined withdrawal of the tool occurs when the shoe of the front tool slide runs along the inclined part of the stationary former, or straight withdrawal when the front tool slide is stationary and the former slide with the former is shifted by a cam to the left so that the shoe drops as the incline of the former runs by.

At the beginning of the cycle, the former slide travels to the right for straight approach of the tool or it is stationary for inclined approach. The tool is fed into the work at the beginning of the working travel as the shoe of the tool slide runs along the upper part of the inclined portion of the former.

Combinations of longitudinal travel of the front tool slide and the front former slide enable various cycles of front tool slide travel to be accomplished, thereby extending the processing capacity of the semiautomatic.

The use of cemented carbide tools for turning steel workpieces is not very efficient due to the insufficient rigidity of the tool slides mounted on round bars and the crowded conditions for continuous chip disposal between the two bars.

It can be seen in the structural block diagram (Fig. 34) that interchangeable drum cams with kinematic contacting have been used as the operative mechanisms of the slides because of the large size of workpiece accommodated and the long slide strokes its machining requires.

The function of the control system extends only to the drive facilities of the semiautomatic and is restricted to the engagement of the high-speed clutch C_2 and the disengagement of the main drive clutch C_1 at the end of the machining cycle. After setting up a new blank between the centres, the clutch for starting the cycle is engaged by hand.

The clutches for the high-speed rotation of the camshaft and for the main drive are controlled by cams mounted on the camshaft.

The high-speed and working feed trains are joined by an overrunning clutch located in the bevel gear of the worm shaft. In setting up their travel, the tool slides are traversed by manual rotation of the worm shaft after disengaging the jaw clutch in the working feed train of the camshaft.

The angle of rotation of the camshaft for working travel of the slides is a constant value ($\alpha_w = \text{const}$), as is the angle of its rotation for the idle travel motions ($\alpha_i = \text{const}$).

The time required for idle travel motions $t_i = \text{const}$; the time required for the working travel t_w is determined by the manufacturing process.

The working feed train of the camshaft drive is set up on the basis of calculations using the kinematic balance equation

$$\frac{\alpha_w}{360} A \frac{z_2}{z_1} = n_{sp}^{1wp} \tag{11}$$

where $\frac{\alpha_w}{360}$ = fraction of a revolution of the camshaft corresponding to the working travel (angle α_w in deg)

A = transmission ratio of the constant transmissions in the train from the camshaft to the spindle

$\frac{z_2}{z_1}$ = transmission ratio of the setting-up device (change gears)

n_{sp}^{1wp} = number of spindle revolutions corresponding to the unoverlapped working travel motions of all the slides.

$$n_{sp}^{1wp} = \sum_{i=1}^{m} \frac{l_i}{s_i} \tag{12}$$

where l_i = length of the working travel of a slide

s_i = feed of the i-th tool slide per spindle revolution.

Only those lengths of working travel are taken into consideration that are not overlapped in time by the travel of other slides.

From equation (11)

$$\frac{z_1}{z_2} = \frac{\alpha_w A}{360 n_{sp}^{1wp}}$$

2-8. Structural and Kinematic Diagrams of Multiple-Spindle Automatics

The kinematic diagram of the model 1240-6 six-spindle automatic bar machine is shown in Fig. 35. The following notation is employed: *1*—four-slot Geneva wheel; *2*—driver of the Geneva wheel; *3*—stock feed cam; *4*—collet closing (stock clamping) cam; *5*—cam for lifting the spindle carrier during its indexing motion; the adjacent cam is for clamping the spindle carrier after it is locked; *6*—spindle carrier locking cam; *7*, *8* and *9*—cross slide cams; *10*—stock stop advancing cam; *11*—duplex cams for actuating the independent-feed tool spindles; *12*—electric motor for turning over the camshaft in setting-up operation; *13*—cam for engaging the high-speed clutch of the camshaft and the brake; *14*—cam for rapid approach of the end tool slide; *15*—cams for engaging and disengaging clutch *17*; *16*—working feed cam of the end tool slide; *17*—clutch for reversing relative rotation of thread-cutting attachment spindle in reference to the work spindles by running the attachment spindle either faster or slower than the work spindles; *18*—clutch for changing over from an automatic cycle to manual controls; *19*—overrunning clutch for joining the working feed and high-speed drive trains of the camshaft; *20*—friction clutch for engaging the high-speed drive of the camshaft; *21*—lubricating pump; *22*—brake; *23*—spline bushing for coupling the two parts of the central shaft; *24*—coolant pump; *25*—screw-type chip conveyer; *26*—electric motor of the chip conveyer drive; *27*—shank for the crank handle used to turn over the spindle carrier in setting up; *28*—stock reel disks.

The structural block diagram of the model 1240-6 automatic is shown in Fig. 36 and its cyclogram, or timing chart, is shown in Fig. 37. The following notation is employed in the two figures: *A*—lubrication control limit switch; *B*—automatic stop of the camshaft (after pressing the FEED STOP push button and when the bar stock has been used up); *C*—spindle carrier indexing; *D*—stock feed; *E*—chucking (stock clamping); *F*—spindle carrier lift; *G*—spindle carrier clamping; *H*—spindle carrier locking; *I*—cross slides in six positions; *J*—stock stop; *K*—working travel of the independent-feed tool spindles in the 3rd and 4th or the 5th and 6th positions, respectively; *L*—rapid approach and withdrawal of the independent-feed tool spindles in the 3rd and 4th or 5th and 6th positions, respectively; *M*—rapid traverse of the end tool slide; *N*—working feed of the end tool slide; *O*—friction clutch and brake for high-speed rotation of the camshaft; *P*—double jaw clutch for a cyclic setup of the thread-cutting attachment spindle in the 3rd and 4th or the 5th and 6th positions, respectively.

Multiple-spindle automatics require high-speed rotation of the camshaft (see Figs. 35, 36 and 37) for the following reasons regardless of the size of workpieces they turn:

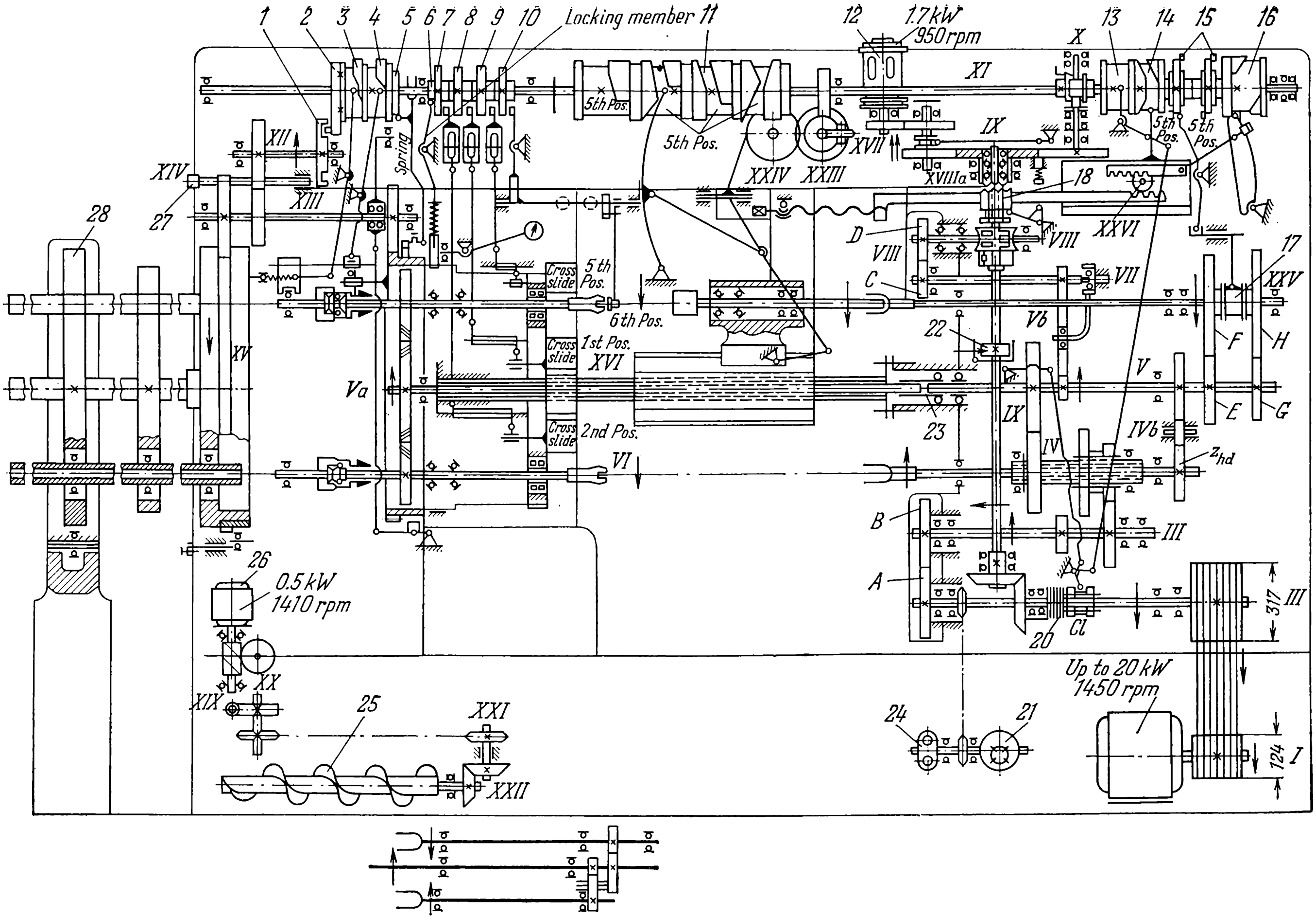


Fig. 35. Kinematic diagram of the model 1240-6 six-spindle automatic

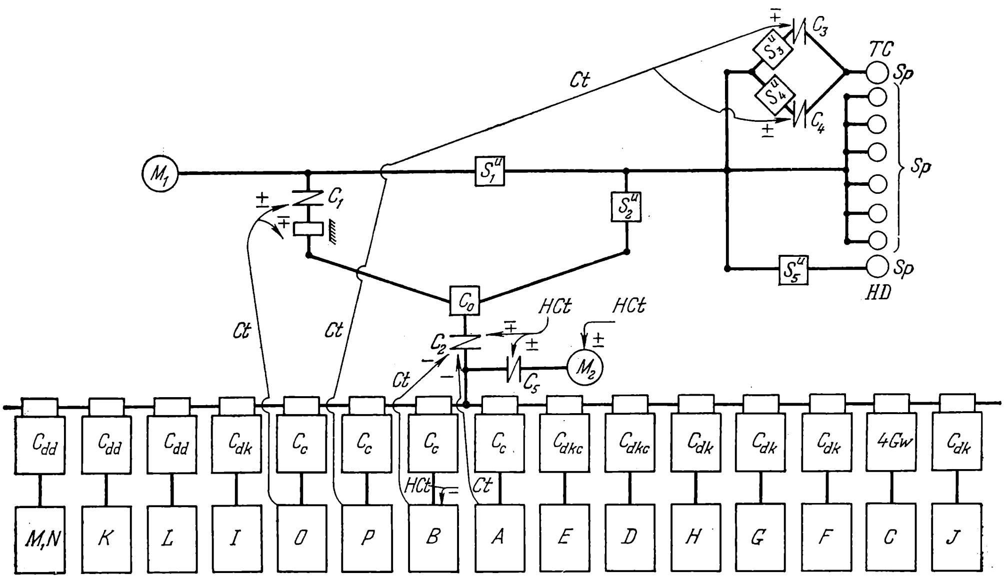


Fig. 36. Structural block diagram of the model 1240-6 automatic: C_{dd}—duplex drum cams; C_{dk}—disk (plate) cams; C_{dkc}—drum cam with kinematic contacting; C_c—control cam; D_c—control disk; $4Gw$—4-slot Geneva wheel; C_0—conjunction of trains of high-speed and working feed rotation of the camshaft by means of an overrunning clutch

1. The end tool slide is usually of hexagonal, square or round cross section with toolholder slots for all positions and is of considerable weight, excluding the use of spring-action contacting in the cam drive of this slide.

2. The indexing cycle, in which the spindle carrier is turned from position to position, requires a large length of withdrawal and approach of both the end tool slide and the holders of the tool spindles sliding along the slots of the end tool slide at an independent rate of feed. For this reason duplex cams with kinematic contacting are used to actuate the end tool slide and the holders of the tool spindles. Such an arrangement requires high-speed rotation of the camshaft during slide approach and withdrawal (see Figs. 20 and 21).

The Geneva wheel mechanism for indexing the spindle carrier operates with a driver angle of rotation of at least 90°. Since the driver is mounted on the camshaft, high-speed rotation is required.

It is evident from the structural block diagram (Fig. 36) that the function of the automatic cycle control system amounts to engaging and disengaging clutch C_1 for high-speed rotation and the brake and for changing over clutches

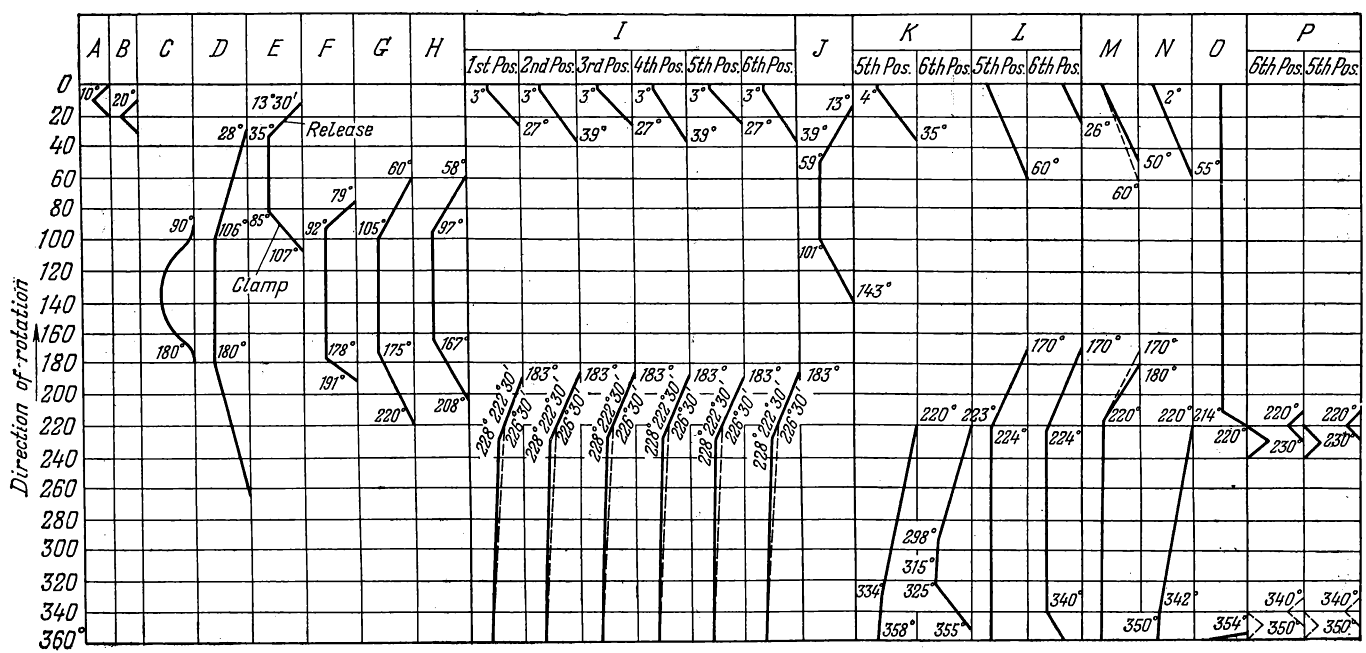


Fig. 37. Cyclogram, or timing chart, of the model 1240-6 six-spindle automatic

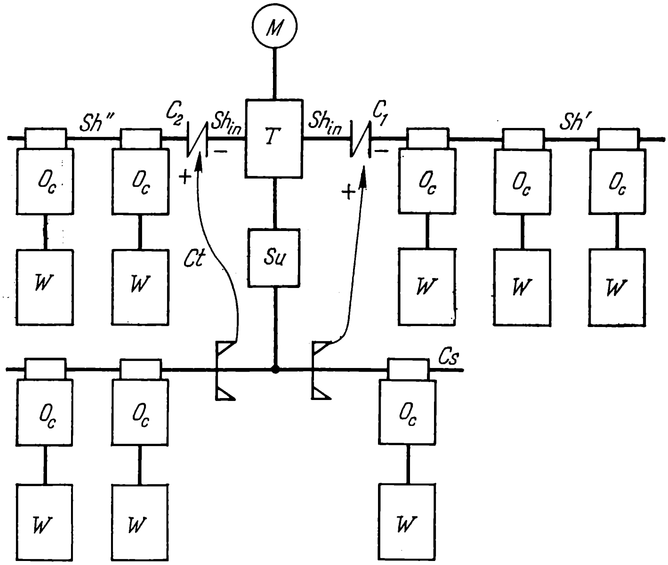


Fig. 38. Block diagram of an automatic machine tool with an auxiliary shaft

C_3 and C_4 in the cyclic setup of the thread-cutting spindle speed so that this spindle runs slower or faster than the work spindles. This is needed in cutting thread and in unscrewing a solid die or tap.

The working feed and high-speed drive trains of the camshaft are joined by an overrunning clutch (C_0 in Fig. 36 and the bushing of worm wheel *19* in Fig. 35).

Also employed in multiple-spindle automatics is a command cam for automatically disengaging rotation of the camshaft. This cam operates a limit switch, disengaging clutch C_2 with a solenoid, if the push button for tentative disengagement of camshaft rotation is pressed, or from a switch in the stock feed mechanism when the piece of bar stock is all used up. In either case, camshaft rotation will stop only after the slides have reached their extreme withdrawn positions.

Thus, the camshaft stops at a definite point in its rotation (see the cyclogram in Fig. 37). This is necessary so as to avoid its re-engagement during spindle carrier indexing which would lead to the shearing of the safety key or safety pin in the drive worm wheel of the camshaft due to the forces of inertia of the camshaft drums and spindle carrier. During regular operation, the inertia of the cam drums relieves the load on the drive during carrier indexing.

Third structural group of cam-controlled automatic machine tools. If one or several operative mechanisms of the working members execute, not one, but several repeated particular automatic cycles during the general automatic cycle of the machine tool, their driving shafts cannot coincide with the main camshaft of the machine. However, if these particular cycles take place simultaneously, the driving shafts of their operative mechanisms can be united into a single common driving shaft (shaft Sh' in Fig. 38), running at a much higher speed than the camshaft and driven by a high-speed auxiliary shaft. This shaft is engaged upon a command from the camshaft by a single-revolution self-disengaging clutch (clutch C_1 in Fig. 38). The clutch is automatically disengaged when the driven common shaft of the operative mechanisms makes one revolution.

Such separation of the operative mechanisms of the working members (as a rule, auxiliary members) into a special group with a common driving shaft engaged by a single-revolution clutch (shaft Sh'' with clutch C_2 in Fig. 38) is also applied to reduce the time required for the operation of the cyclic operative mechanisms of the auxiliary working members.

A classical example of an automatic machine tool of the third structural group is the single-spindle automatic screw machine (Figs. 39 and 40) in which there are six particular change-over cycles (Cy_2 in Fig. 41) of turret operation (Fig. 42) during the general automatic cycle for machining one workpiece (during one revolution of the camshaft).

The operation at each turret position consists of three cycles of auxiliary (handling) motions:

1. Withdrawal and approach (Cy_2''' in Fig. 41) of the turret slide by means of the crankgear mechanism.

2. Unlocking and locking (Cy_2'' in Fig. 41) of the turret with the locking pin actuated by a single-edge cylinder cam.

3. Turret indexing through $\frac{1}{6}$ revolution (Cy_2' in Fig. 41) by means of a Geneva wheel mechanism.

The crank, cylinder cam and driver of the Geneva wheel are mounted on a common shaft (Fig. 42) which is driven by an auxiliary shaft (with a speed of 120 rpm) through a single-revolution clutch (see Fig. 40).

The particular working cycles of the front, rear and vertical cross slides, as well as the turret slide, are accomplished by plate cams with spring-action contacting (see Fig. 41). This enables the general machining cycle to be accomplished at constant speed of the camshaft.

The provision of a crankgear mechanism for rapid withdrawal and approach of the turret during indexing limits the function of the turret slide lead cam to the working travel and withdrawal of the turret along the drop curve of the cam to the position at the beginning of the next working travel motion.

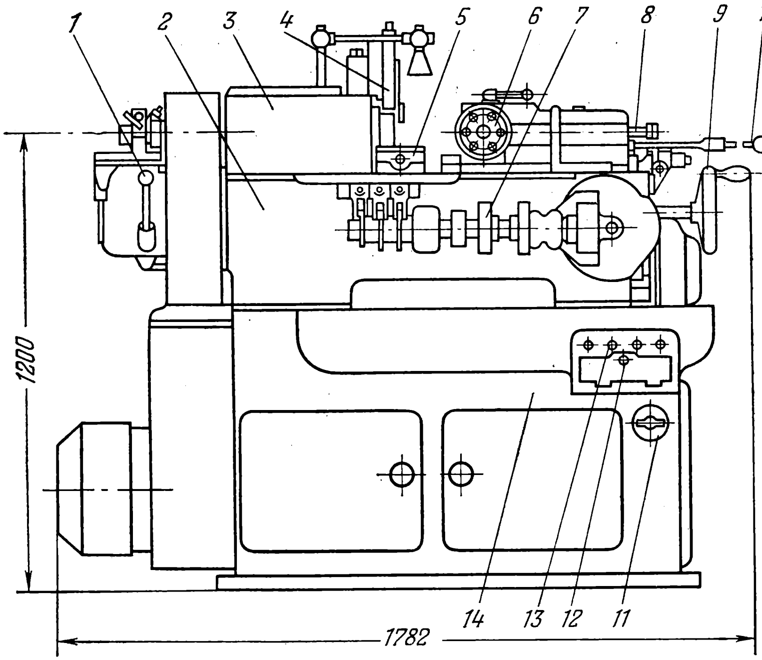
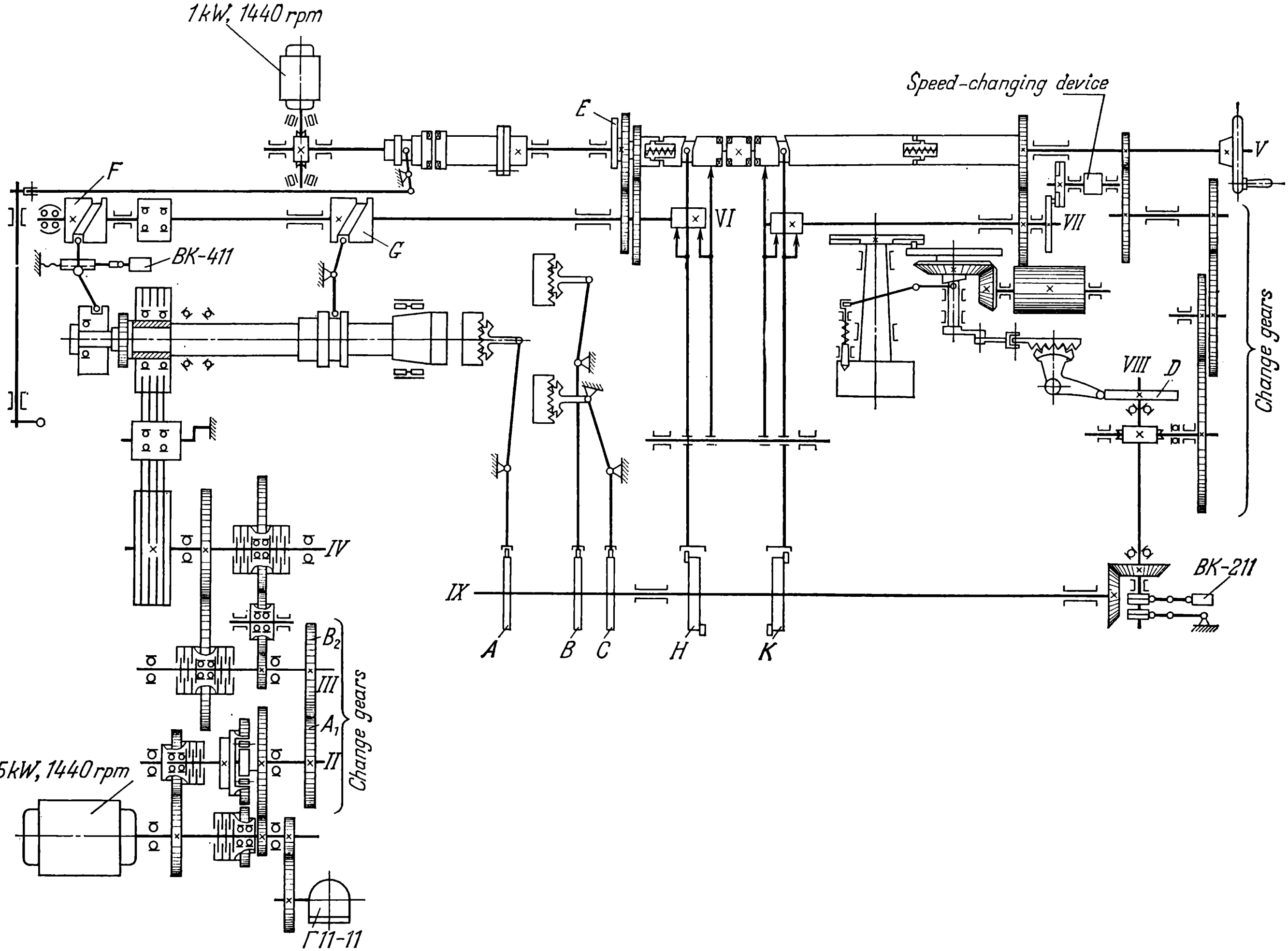


Fig. 39. General view of the automatic screw machine, model 1Б136:
1—lever for engaging the clutch of the auxiliary shaft drive; *2*—bed; *3*—headstock; *4* and *5*—vertical and horizontal tool slides, respectively; *6*—turret; *7*—camshaft; *8*—adjustment for positioning the turret slide in reference to the spindle nose; *9*—handwheel for rotating the auxiliary shaft and the camshaft; *10*—lever for traversing the turret slide by hand; *11*—main line rotary switch; *12*—control panel for setting up the spindle speeds; *13*—push button controls of the spindle drive electric motor; *14*—base

As a result, the turret slide lead cam can provide up to six working travel motions during the machining cycle, including approach of the stock stop.

The common shaft of the cams for chucking and stock feed is driven by the auxiliary shaft through a single-revolution clutch.

The function of the control system in machines of this structure (see Fig. 41) is limited to the control of the transmission mechanisms: engagement of the single-revolution clutches by command cams on the camshaft, cyclic reversal (R_c) of spindle rotation when a cam of the camshaft operates limit switches, and cyclic changes of the spindle speeds by means of a special change-over switch which is turned to six consecutive positions, corresponding to turret positions (see Fig. 40), by a special Geneva wheel mechanism linked to the turret indexing drive.



1 kW, 1440 rpm
Speed-changing device
E
F
G
BK-411
VI
VII
VIII
V
D
Change gears
IX
A
B
C
H
K
BK-211
IV
III
II
B2
A1
Change gears
45 kW, 1440 rpm
Г11-11

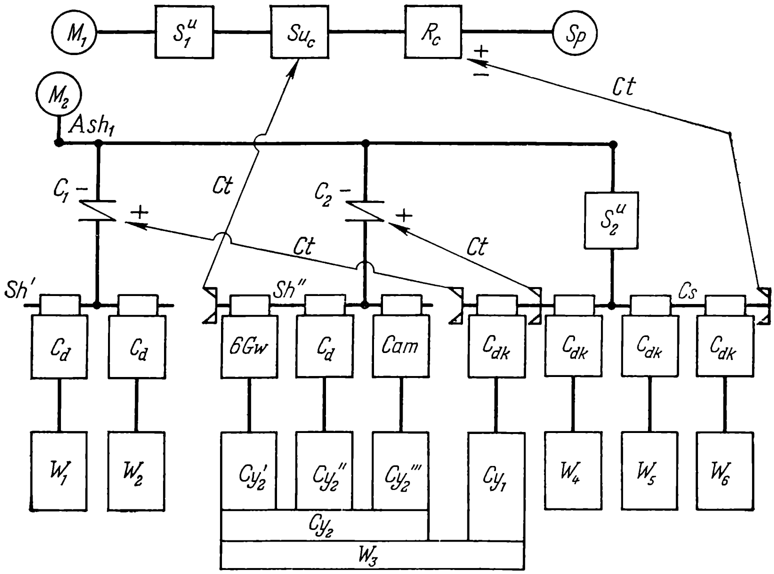


Fig. 41. Structural block diagram of the model 1Б136 automatic screw machine: ASh_1—auxiliary shaft; Sh' and Sh''—driving shafts of the groups of operative mechanisms; C_1 and C_2—single-revolution clutches; Su_c and R_c—cyclic setting-up facilities and cyclic spindle reversal; $S^u{}_1$—setting-up facilities in the spindle drive; $S^u{}_2$—setting-up facilities in the camshaft drive; W_1—stock feed; W_4, W_5 and W_6—cross tool slides; W_2—collet operation (stock clamping); W_3—turret slide; Cy_1—working feed and return of the turret slide; Cy_2—turret indexing; Cy_2'''—approach and withdrawal; Cy_2''—unlocking and locking; Cy_2'—turret rotation; C_{dk}—disk (plate) cams with spring-action contacting; C_d—drum cams

The general cyclic co-ordination of the particular cycles of the main (shaft *Cs* in Fig. 41) and auxiliary (shafts *Sh'* and *Sh''* in Fig. 41) working members is accomplished by the control system which engages the rotation of the driving shafts of the operative mechanisms of the auxiliary members at definite stages in the general automatic cycle.

Since the single-revolution clutches C_1 and C_2 are engaged by kinematically rigid (positive) mechanisms, the linkage between the main and auxiliary working members remains kinematic during the whole automatic machining cycle.

The construction of the turret slide of the model 1Б136 automatic screw machine is illustrated in Fig. 43.

The turret indexing cycle (Fig. 44) proceeds in the following manner.

Beginning of indexing (Fig. 44*a*). The roll of the segment gear runs down along the drop curve of the cam; under the action of the spring (spring-action contacting in the cam mechanism) the slide travels to the right. The single-

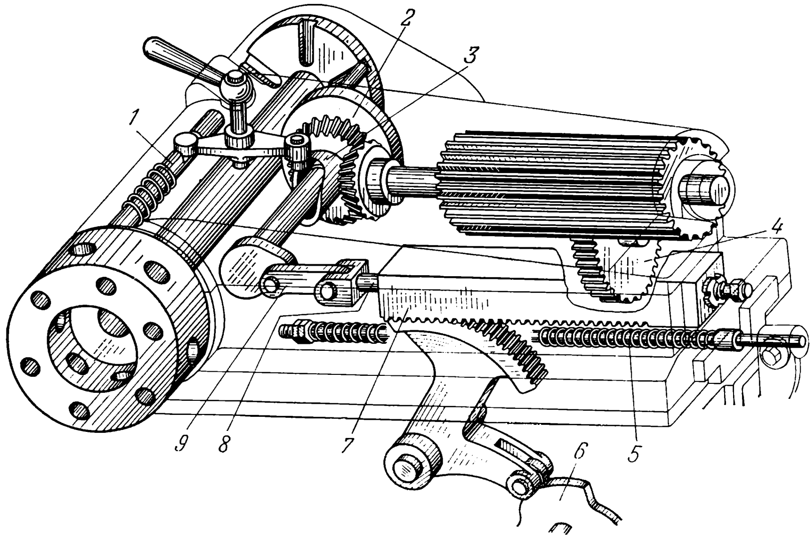


Fig. 42. Principle of the turret slide:

1—locking pin; *2*—Geneva wheel driver; *3*—single-edge cylinder cam of the locking pin; *4*—gear of the drive from the auxiliary shaft; *5*—spring for spring-action contacting in the cam mechanism for turret slide feed; *6*—turret slide lead cam; *7*—rack sliding in the turret slide bracket housing; *8*—connecting rod; *9*—crankgear mechanism for rapid withdrawal and approach of the turret slide, linked through connecting rod *8* to the rack

revolution clutch for turret indexing is engaged; the crank begins to rotate, the distance between A and B decreases and the speed of withdrawal increases. The cam mechanism remains in contact.

End of turret slide withdrawal to the stop (Fig. 44*b*). The force of the spring acts against the stop. Spring-action contacting ceases in the cam mechanism.

Beginning of turret rotation (Fig. 44*c*). Upon further rotation of the crankshaft, the rack travels forward (to the left), turning the segment gear so that the roll moves away from the cam. The driver roll enters the slot of the Geneva wheel. The locking pin is fully withdrawn from its socket.

End of turret rotation (Fig. 44*d*). The crank passes the dead-centre position, the rack begins to travel back (to the right), the roll approaches the cam, the driver roll leaves the Geneva wheel slot and indexing is completed. The locking pin re-enters its socket.

Beginning of turret slide approach by means of the crankgear mechanism (Fig. 44*e*). Upon further rotation of the crank, the rack continues to travel to the right until the roll of the segment gear contacts the surface of the turret slide cam. As the crank continues to rotate, the distance between A

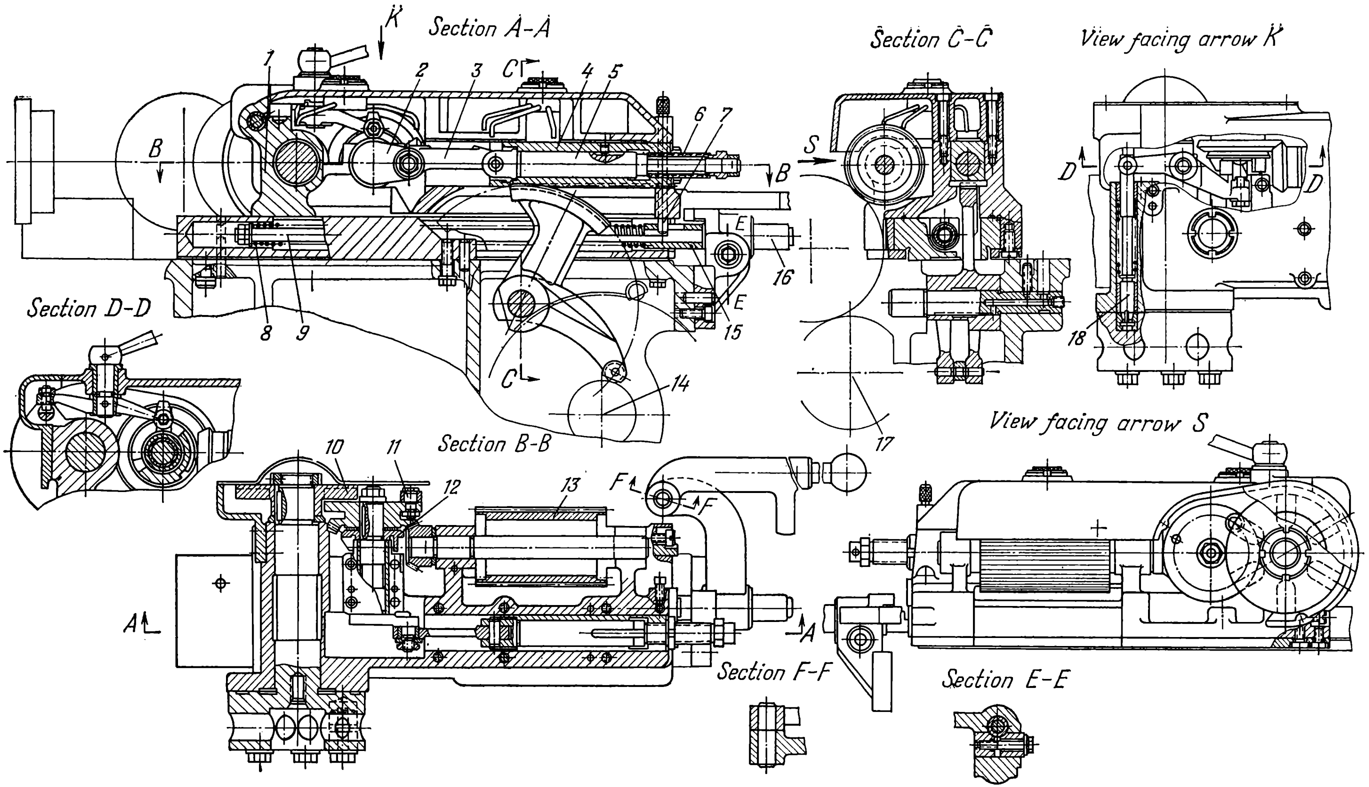


Fig. 43. Turret slide of the model 1Б136 automatic screw machine:

1—turret slide bracket; *2*—crank; *3*—connecting rod; *4*—hollow rack sliding in a slot of the bracket; *5*—rod secured in the rack; *6*—threaded bushing for clamping and adjusting the position of rod *5* in rack *4* in setting up the turret slide; *7*—pin press-fitted into the bracket; the end of this pin enters a slot in rod *9*; *8*—spring; *9*—rod with a head at the right-hand end bearing against the end face of sleeve *16*; *10*—turret Geneva wheel; *11*—driver roll of the Geneva wheel; *12*—single-edge cylinder cam for actuating the locking pin; *13*—input spool gear of the turret indexing mechanism; *14*—camshaft axis; *15*—sleeve with slot; *16*—sleeve-stop; *17*—auxiliary shaft axis; *18*—turret locking pin

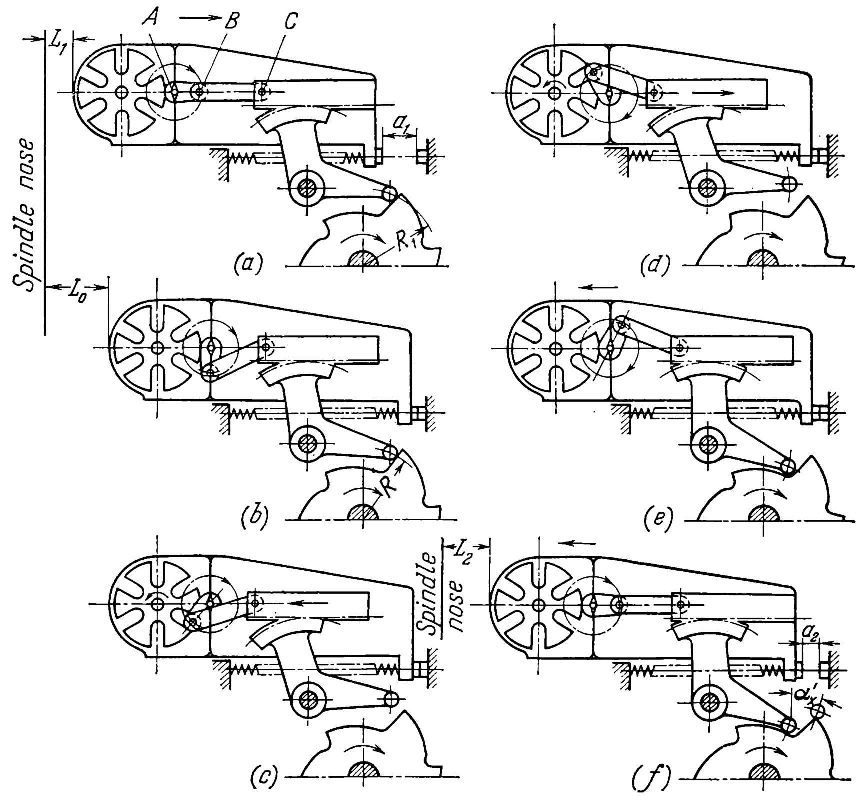


Fig. 44. Steps in turret indexing in an automatic screw machine

and B increases with the cam mechanism in contact, the slide leaves the stop and is rapidly advanced to the machining zone.

Simultaneously, the turret slide is withdrawn slightly as the roll runs along a drop curve of the cam.

End of indexing (Fig. 44*f*). Slide approach is completed when the crank is in its rear dead-centre position. At this moment, the single-revolution clutch is disengaged and locked and, with it, all the gears of the indexing mechanism and the shaft of the crankgear mechanism. At the end of turret indexing, the roll should be at the end of the drop curve on the lead cam and at a radius which is 1.5 mm less than the radius corresponding to the beginning of the next working travel motion of the turret slide. If indexing ends when the roll has not reached the end of the drop curve but is at a larger radius, the slide

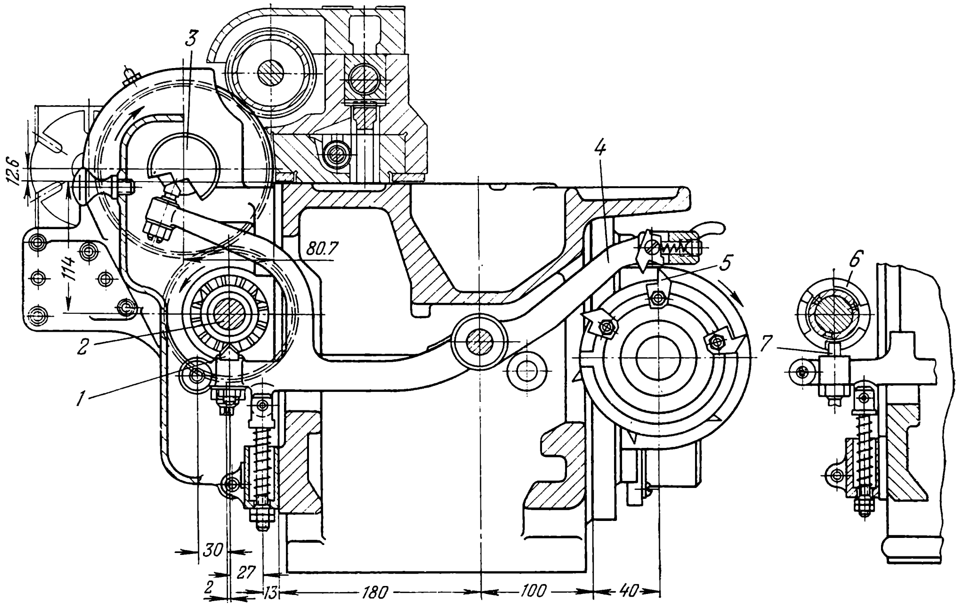


Fig. 45. Mechanism for engaging the single-revolution clutch for indexing the turret in automatic screw machines, models 1A136 and 1Б136

will be advanced after indexing farther forward than the beginning of the next working travel motion, and the tools will run up against the work. Then the slide will be withdrawn to the end of the drop curve on the cam and will start moving forward again at the beginning of the next working travel motion.

For this reason, the command cams of the single-revolution clutch for turret indexing are adjusted to end the indexing cycle so that no return motion of the slide occurs after indexing.

At the end of turret slide withdrawal (Fig. 44*b*) sleeve *15* (Fig. 43) runs up against sleeve-stop *16*, transmitting to it the force of spring *8*. The crank pulls the rack into the housing of the turret slide bracket, and the slide continues to travel by inertia. At this, pin *7*, moving along the slot in sleeve *15*, runs up against the end of the slot in rod *9*, pushing this rod to the right. Rod *9* compresses spring *8*, thereby absorbing the kinetic energy of the slide. After this, the spring moves the slide to the left and pin *7* bears against the left edge of the slot in sleeve *15*.

Single-revolution clutches. Cam-type single-revolution clutches are used in automatic screw machines to engage the drive of single-revolution opera-

tive mechanisms from the auxiliary shaft (see clutches C_1 and C_2 in Fig. 41). These clutches automatically disengage after one revolution of the input shaft of the cyclic operative mechanism, i.e., after one particular cycle of motion of the working member.

In automatic screw machines, models 1A136 and 1Б136, auxiliary shaft *2* (Fig. 45) makes $1\frac{1}{3}$ revolutions during a turret indexing cycle. Cam *3* on the intermediate shaft, which makes one revolution during this cycle, allows pin *7* to enter the shaped recess *6* of the clutch member, disengaging the clutch at the end of the turret indexing cycle. Since the clutch member makes $1\frac{1}{3}$ revolutions during the indexing cycle, it has three shaped recesses *6* and three slots for locking pin *1*.

Upon rotation of cam *3*, its bevelled surface imparts an additional rocking motion to lever *4* which raises the latch at the end of the lever above the apex of command cam dog *5*, allowing the latch to turn by spring action so that it does not run against cam dog *5* when pin *7* enters recess *6*.

CHAPTER 3

AUTOMATIC MACHINE TOOLS WITH LEAD-SCREW-DRIVEN WORKING MEMBERS

3-1. Structural Properties of Lead-Screw-and-Nut Pairs

The screw-and-nut pair constitutes a noncyclic operative mechanism of the main working members in an automatic machine tool. Noncyclic operative mechanisms are characterized by the invariability of the kinematic linkage between the input (driving) shaft of the operative mechanism and the working member. This linkage is expressed by the magnitude of working member displacement per revolution of the driving shaft of the operative mechanism, i.e., the "pitch" of the traversing device of the operative mechanism. This device serves to convert rotary motion of the driving (input) shaft into translatory motion of the working member.

Noncyclic mechanisms of the working members in automatic machine tools may be of the lead-screw-and-nut and the rack-and-pinion types. These mechanisms are not of the single-revolution type. The number of revolutions of the driving shaft during a particular cycle of a working member may vary in accordance with the length of travel of the working member and the pitch of the operative mechanism.

Noncyclic operative mechanisms (chiefly of the lead-screw-and-nut type) are used in automatic machine tools to actuate the main working members; they are less often used for the auxiliary members (in the operating devices of chucks).

A small "pitch" of the traversing device is typical of lead screws which consequently require only a small degree of reduction so that no worm gearing is needed in the working feed drive. Thus

$$i = \frac{s}{t} \tag{13}$$

where t = lead (pitch × number of starts) of the screw, mm
s = feed, mm, per revolution of the spindle
i = transmission ratio of the working feed train from the spindle to the lead screw.

The low value of the torque on the lead screw is due to the small pitch [see equation (104) and Sec. 6-4 in Vol. 3].

Characteristic of screw-and-nut pairs is their small radial overall size. Therefore, a lead screw can transmit heavy traversing forces without being subject to a high torque or contact stresses. A lead screw can be used for a long length of travel of the working member. A lead screw drive ensures smooth feed motion (see Sec. 6-4 in Vol. 3).

If the lead-screw nut rotates, it is possible to design the conjunction of the working feed and rapid traverse trains in the screw-and-nut pair. It is also possible to add the motions of the power working feed produced by screw rotation to manual feed produced by simultaneously rotating the nut by hand.

3-2. Structure of Lead-Screw-Driven Automatic Machine Tools

Of prime structural significance is the constant pitch of the lead screw, as a result of which the lead-screw-and-nut pair (in the same way as the rack-and-pinion pair) constitutes a noncyclic operative mechanism. Such mechanisms are characterized by the constant kinematic linkage between the input shaft of the operative mechanism and the working member.

Consequently K_c—the cyclic change in the kinematic linkage between the common transmission mechanism T of the machine tool and the working member W, needed to obtain the required cycle of motions of the working member—is accomplished by cyclic setting-up and reversal of the drive from the common transmission mechanism to the lead screw of the given working member (see Fig. 46).

The character of the cyclic setups for producing K_c depends upon the cycle of motions of the working member. Efforts are made to keep this cycle as simple as possible, for example: rapid approach, working travel at one rate of feed and rapid withdrawal at the same speed as the approach. This enables the cyclic setup to be restricted to starting, stopping and reversing the rapid traverse drive motor by means of limit switches. Electromagnetic clutches are also employed to switch over the transmissions of the drive. Mechanical controls are rarely used for this purpose.

As mentioned above, a lead screw is a noncyclic operative mechanism. The particular automatic cycle of its working member is executed, not in one, but in several revolutions of the lead screw. Thus, the lead screw cannot coincide with the common camshaft of the machine tool which makes one revolution during the general cycle of the machine.

Under these conditions, the particular cycle of motions of the working member cannot be controlled in a centralized manner from a camshaft, using a time-sequence control system. Hence, the cyclic setups of the particular transmissions in the drive to any lead screw are controlled, at various stages of its particular cycle of motions, by the travel of the working member (in-travel controls) either directly or through some device (master switch)

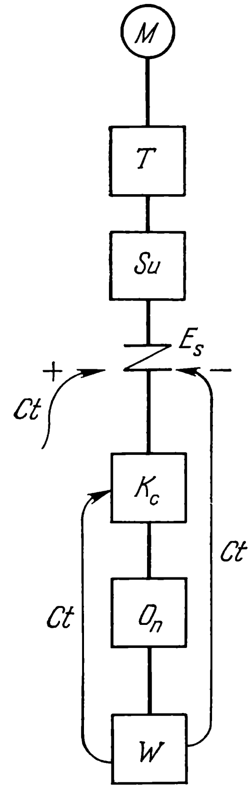


Fig. 46. Block diagram of a driven working member with a noncyclic operative mechanism:

M—motor; T—transmission mechanism of the machine tool; S_u—setting-up facilities for the working feed drive train; $\pm E_s$—device for engaging and disengaging the operative mechanism drive; K_c — cyclic re-engagements of the kinematic system of the particular drive of an operative mechanism to obtain its automatic cycle of motions; O_n—noncyclic operative mechanism; W—working member; Ct—function of the control system of the automatic cycle

linked kinematically to the working member or the lead screw.

In distinction to cyclic operative mechanisms, at the beginning of the particular cycle of motions of a working member a noncyclic mechanism requires a device E_s for engaging the particular drive of the lead screw of the given working member (Fig. 46) at a definite stage of the cycle (or cycles) of some member (or group of members). This device co-ordinates the particular cycle of a working member with the general automatic cycle of the machine tool.

The disengagement of E_s of the particular drive of the operative mechanism is accomplished by the working member itself at the end of its cycle of motions (Fig. 46).

Thus, in using a lead screw or other noncyclic operative mechanisms, the general cyclic interlinkage and the co-ordination of the cycles of the various working members of the machine tool are accomplished by means of in-travel control of the general automatic cycle and with mutual interlinkage and interlocking of the particular systems for controlling the cycles of the various working members.

From the representative block diagram (Fig. 47) it is evident that the functions of the system for control of the automatic cycle in a machine tool are considerably complicated when lead screws and other noncyclic operative mechanisms are used. The control system is made more involved, not only by the general cyclic linkages between the working members, but by the control of the particular drives of the operative mechanisms of the working members to obtain their cycles of motion.

It has been indicated above that a lead-screw-and-nut pair is rarely employed as the operative mechanism of the auxiliary working members (except as the operating devices of chucks). At the same time, it should be noted that in using lead screws for operative mechanisms of the main working members, the co-ordination of the particular cycles

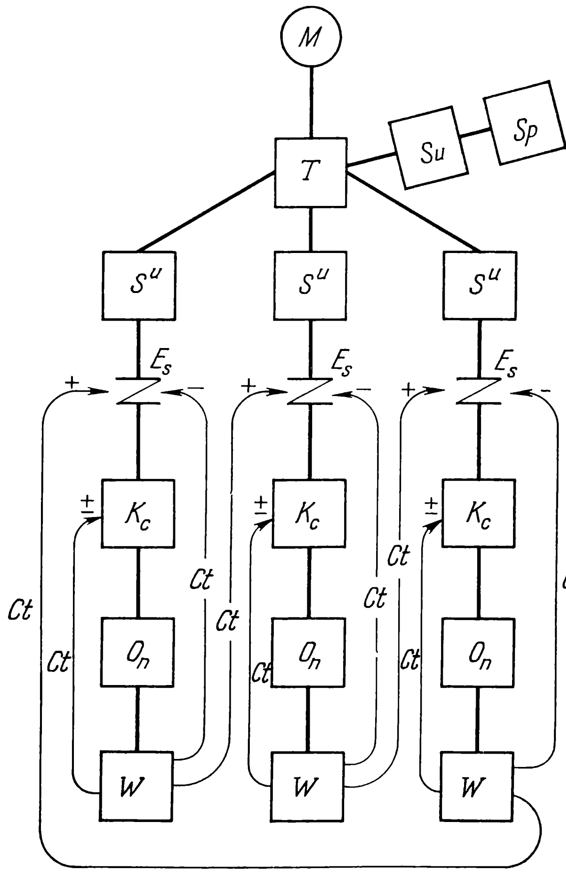


Fig. 47. Block diagram of an automatic machine tool with noncyclic operative mechanisms

in the general automatic cycle by means of the control system enables various types of operative mechanisms (mechanical cyclic, hydraulic and pneumatic) to be employed for the auxiliary working members of a single machine tool. This can be more easily done with an electrical command circuit in the control system.

3-3. Control System of the Automatic Cycle with the Application of Lead Screws

When lead screws or other noncyclic operative mechanisms are employed, the functions of the control system are considerably extended in comparison with cam-controlled automatic machine tools. The engagement and disengagement of all the working members, the switching over of the transmissions and electric motors to make up the cycle of motions of each main and auxiliary working member are all executed by means of the control system. As a result, the amount of contacting electrical apparatus is increased and the reliability of machine tool performance is reduced in comparison with cam-controlled automatic machine tools.

As mentioned above, an in-travel control system is used for the working members when lead screws are incorporated in the design.

The advantages of this system are that it excludes the possibility of cycle disadjustment in space and requires no involved preparatory and assembling operations to change over the machine tool to handle a new workpiece. A machine with such a control system can be changed over in less time than a cam-controlled automatic machine tool. For this reason, lead screws are used not only for lengths of travel of the main working members over 300 mm (the maximum permissible length of travel for drum cams), but in machine tools with shorter lengths of travel but designed for lot production.

The setting-up of the drive for the lead screw of the working member, to impart the required cycle of motions, leads to discontinuities in the kinematic trains in the drive, to transient processes and to mechanical and electrical slippage. As a result, a less rigid kinematic linkage is obtained, and the lengths of travel and the time required are less exactly limited than in cam and other cyclic operative mechanics. This makes lead screws and other noncyclic operative mechanisms inconvenient for machine tools in which the cycles of the main working members are very short, i.e., for machining small workpieces.

The decentralized nature of the control system makes it more difficult to set up. This drawback can be alleviated to some extent by the use of master control switches in which the setting-up of the particular automatic cycles of the various main working members is concentrated. The master control switch is kinematically linked to the working member. If there are several main working members, several master switches are used, separate ones for each member.

In-travel control systems are also employed for auxiliary members.

These particular control systems of the various working members are integrated by mutual cyclic linkages with interlocking features into the general control system of the automatic cycle.

3-4. Structural and Kinematic Schemes

Automatic and unit-built machine tools, employing lead screws as the drives of the working members, can be classified into three groups depending upon the motions of the screw and nut:

(1) rotating nut and fixed screw;

(2) rotating screw and nonrotating nut;

(3) rotating screw and nut.

An example of a device of the first group can be found in a tapping power unit (Fig. 48). This unit has a simple cycle of motions: working travel forward—return (tap withdrawal) at high speed, using a two-speed electric motor in the main drive. The feed drive has a positive kinematic linkage with the spindle drive. The power unit operates on a semiautomatic cycle controlled by trip dogs which are clamped on the housing of the unit and actuate limit switches located in box *6*.

Drive in a Guide Plate

In power units where the working feed motion is of small length, it is accomplished by a flat cam which traverses the spindle quill. If, in this case, long approach and withdrawal of the power unit are required, an independent drive of the lead screw is used. This mechanism is located in the guide plate (Fig. 49).

At the end of the rapid approach, one of the trip dogs of the power unit operates a limit switch, switching off motor *1* and de-energizing solenoid *7*. Spring *9* engages jaw clutch *4*, linking the screw drive with the drum of shoe brake *3* whose braking torque is substantially higher than that of friction clutch *2*. The screw stops immediately after the jaw clutch is engaged, while electric motor *1* is braked by the friction braking clutch and stops after a short time without causing overtravel of power unit *5* due to run-out of the rotor.

In respect to the rotation of the screw and nut, semiautomatic multiple-tool lathe, model 1730 (Fig. 50), is of intermediate design. The semiautomatic working cycle is carried out without rotation of the nut, while in setting up the lathe, the carriages are traversed by hand by rotating the nut. This has affected the construction of the nut and its rotation drive, but the kinematic layout is the same as for a nonrotating nut.

The cross-feed carriage is driven from the longitudinal carriage (Fig. 51) and they operate simultaneously. The cycle of carriage travel is not complex. Tool relief (retraction) at the end of the working feed and tool advance after carriage withdrawal are provided by the action of the cam bar (see Fig. 54).

Cyclic re-engagement in the drive of the carriages is performed by switching over the rapid traverse motor M_2 and clutch C (Fig. 51) with a reversible

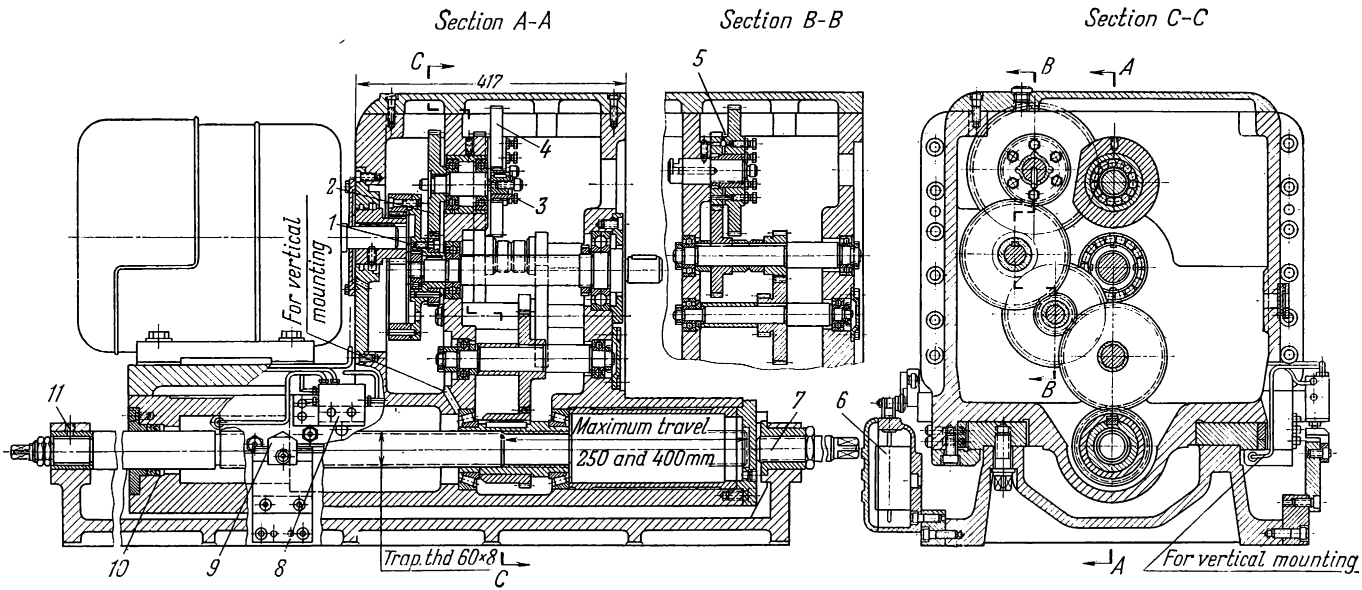


Fig. 48. Tapping power unit with feed effected by a rotating nut traversing along a fixed lead screw:
Feeds are set up with change gears *1-2* and *3-4*; *5*—safety clutch; *6*—limit switch box; *7*—safety stop screw; *8*—lubricating pump; *9*—trip dog; *10*—packing; *11*—safety shear pin

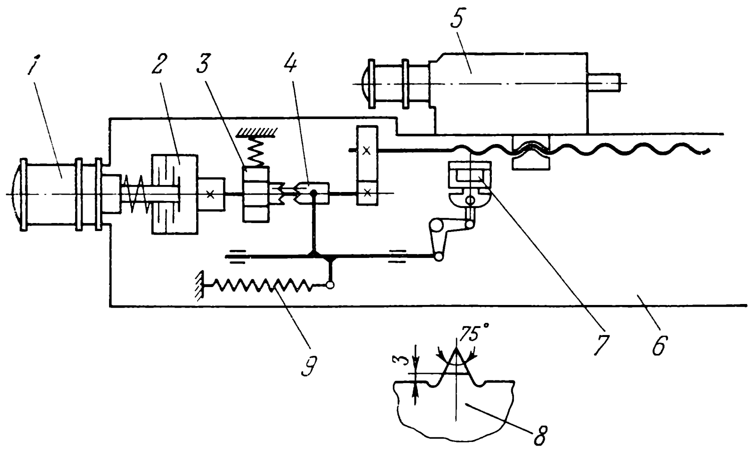


Fig. 49. Diagram of a drive in a guide plate:

1—guide plate drive motor; *2*—braking clutch; *3*—shoe brake; *4*—jaw clutch for engaging the brake; *5*—power unit with flat cam control; *6*—guide plate; *7*—solenoid for disengaging the jaw clutch; *8*—tooth profile of clutch *4*; *9*—spring for engaging clutch *4* of the brake

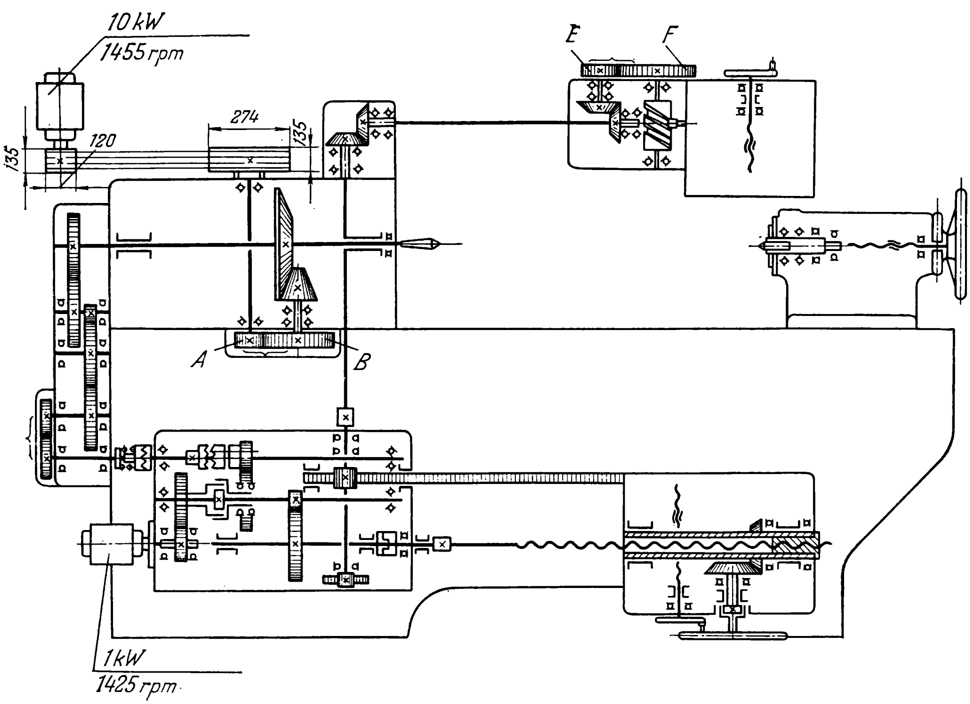
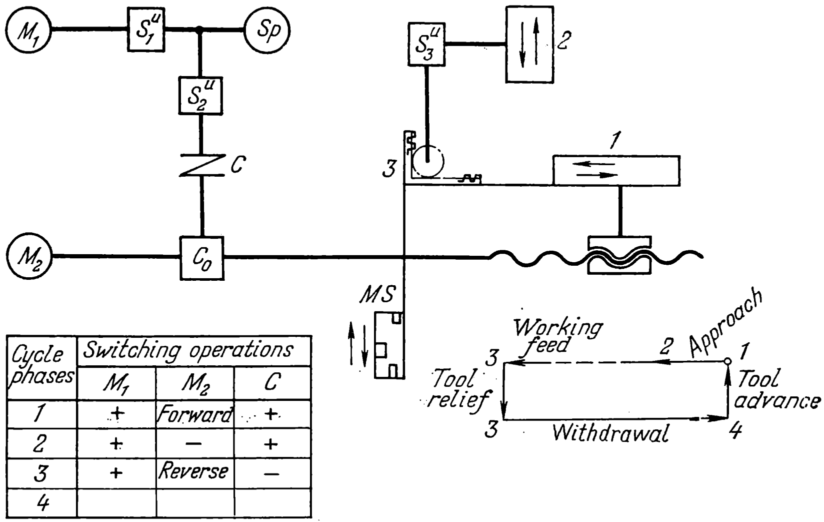


Fig. 50. Kinematic diagram of the semiautomatic multiple-tool lathe, model 1730



Cycle phases	Switching operations		
	M_1	M_2	C
1	+	Forward	+
2	+	—	+
3	+	Reverse	—
4			

Fig. 51. Structural diagram of the model 1730 lathe:

M_1—main drive motor; M_2—rapid traverse motor; Sp—spindle; S_1^u—setting-up facilities for the spindle drive; S_2^u—setting-up facilities for the feed drive; S_3^u—setting-up facilities for cross carriage feed; *1*—longitudinal carriage; *2*—cross-feed carriage; *3*—rack-and-pinion drives to the master control switch MS; C_0—junction of the trains for rapid traverse and working feed of the carriages (reversible single-direction overrunning clutch); C—clutch for engaging the feed drive

single-direction overrunning clutch C_0 (Fig. 52) which is incorporated in the design. The main drive motor is switched off and on in connection with the removing of the finished workpiece and setting up the next blank.

The master control switch is linked to the feed gearbox (Fig. 52) which is connected to the working feed drive through toothed safety clutch *1*. Toothed clutch *2* engages the working feed train when cradle *3*, mounted on shaft *13*, is raised by shifting lever *12*. The reversible overrunning clutch serves to join the working feed train to the rapid traverse train which is powered by a reversible electric motor.

Rack *7*, mounted in the longitudinal carriage, rotates a pinion integral with shaft *8* which transmits rotation through spline coupling *9* to the feed gearbox of the cross-feed carriage. Mounted on the other end of shaft *8* is gear *6* which transmits motion to the rack of the master control switch. This rack carries the trip dogs (stops).

Cradle *3* is retained in the raised position by locking member *5*. This member is retracted from its seat *4* when lever *12* is shifted downward about the axis of the cradle, or when cradle lever *10* with the screw-stop is shifted about

the same axis by one of the trip dogs actuating roll *11* mounted in the body of the lever.

The longitudinal carriage (Fig. 53) of the semiautomatic lathe has a cam bar *6* with an inclined edge along which the carriage is withdrawn at an angle during rapid return and along which angular infeed is performed to a depth up to 10 mm at the end of the rapid longitudinal approach movement of the carriage.

At the end of the working feed, the tool relief mechanism (Fig. 54) retracts the tools 1 mm from the work. This occurs when bar *7* runs up against left-hand stop *6*, and is shifted to the right in respect to bar *3*. At the point where the projections of one bar coincide with the recesses of the other, spring *7* (Fig. 53) retracts bars *3* and *4* (Fig. 54), roll *2*, its pin and the cross slide with the tools from the work. At the end of the rapid return, bar *7* runs up against the right-hand stop *6*, "cocking" the bars into the position shown in Fig. 54. As a result, bar *4*, roll *2* and the cross slide are shifted with the tools toward the work. This is the tool advance movement.

The front carriage is traversed manually by the rotation of the lead screw nut through gearing when handwheel *16* (Fig. 53) is turned. Sleeve *13*, keyed to the handwheel shaft, has projections which force the rollers *14* out of the wedged positions (down into the recesses of cam *12* of the two-direction reversible overrunning clutch). Cam member *12* is keyed on the hub of bevel gear *15*.

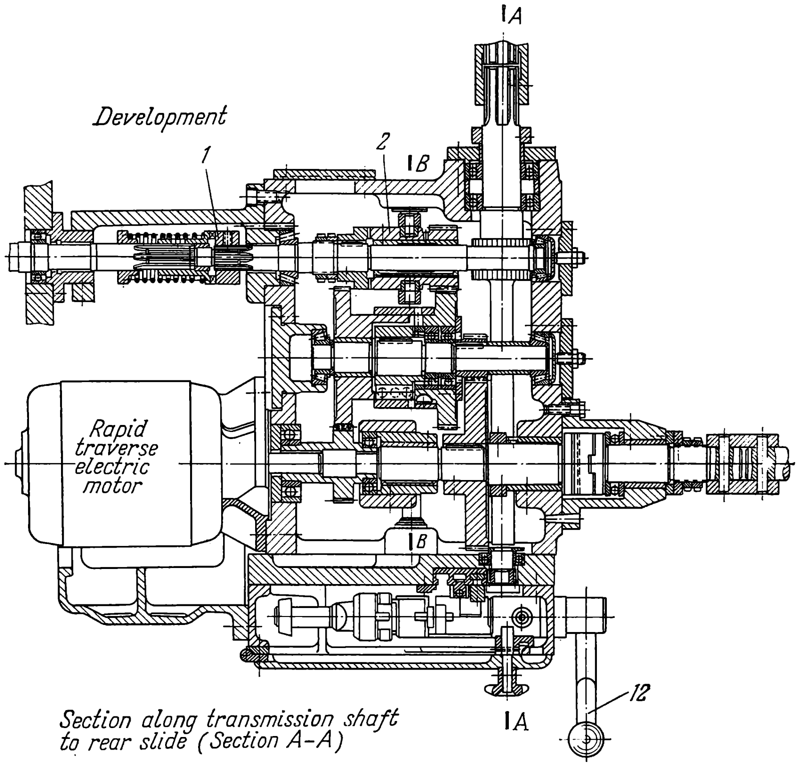
Upon operation from the lead screw, rotation of the nut is braked by overrunning clutch *12*, two rollers *14* of which are wedged between the cam member and the stationary housing *11*.

Semiautomatic Cycle of the Lathe

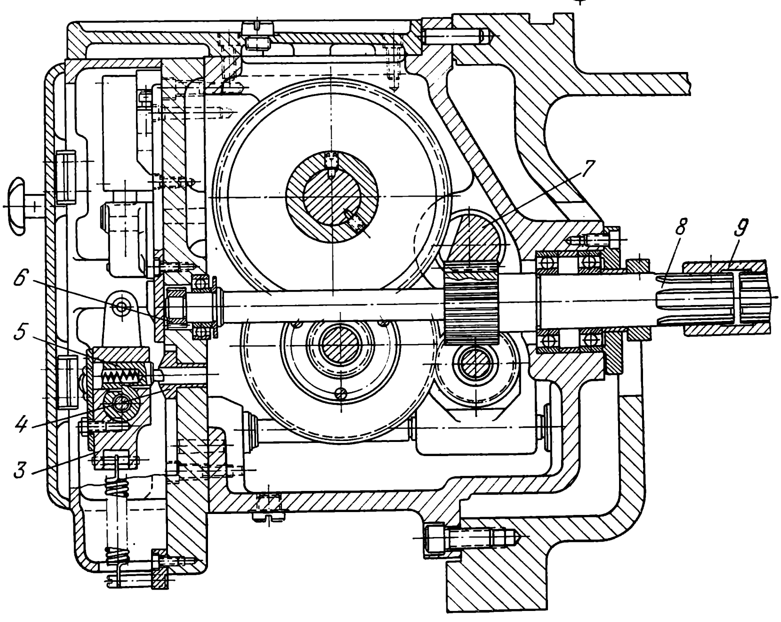
Initial position. Rack *3* is in the upper position (Fig. 55) while the cradle is in position B. Through lever *2*, trip dog *1* holds down the pin of limit switch LS_3 so that contacts 1_a-*2* are closed and contacts *1-5* are open (Fig. 56). Trip dog *9* of the cradle (Fig. 55) holds down the pin of limit switch LS_2 so that its contacts *6-7* (Fig. 56) are closed and its contacts *1-5* are open. The normally open limit switch LS_1 is not tripped.

Rapid approach of the carriages. When the cradle is raised by shifting the control lever, screw-stop *8* (Fig. 55) of lever *7* in the cradle presses the pin of limit switch LS_1, while trip dog *9* of the cradle releases the pin of limit switch LS_2, opening its contacts *6-7* and closing its contacts *1-5* (Fig. 56). The coil of contactor *CRF* is energized along circuit *1*, 1_a, *2*, *3* and *4*, and the main contacts of *CRF* are closed, switching on the rapid traverse motor *RM* for forward rotation. At the same time, intermediate relay *IR-1* is closed, thereby closing relay *IR-2* and the contacts of contactor *CM*. This switches on the main drive motor *MM*, coolant pump motor *PM* and energizes sole-

Development



Section along transmission shaft to rear slide (Section A–A)



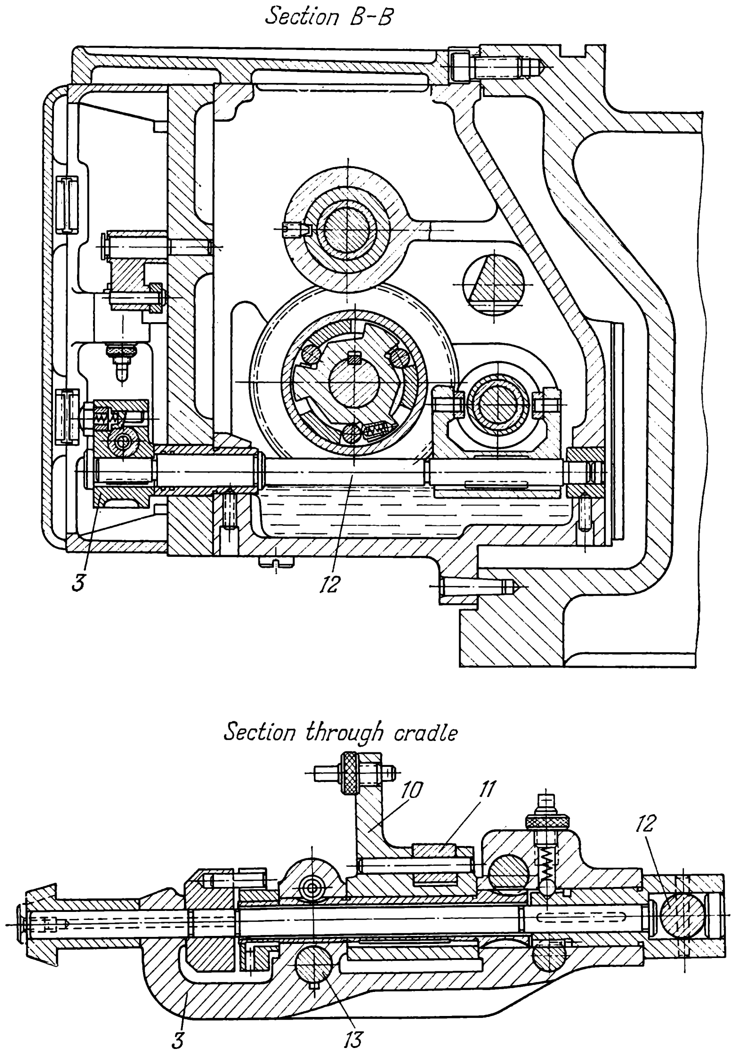
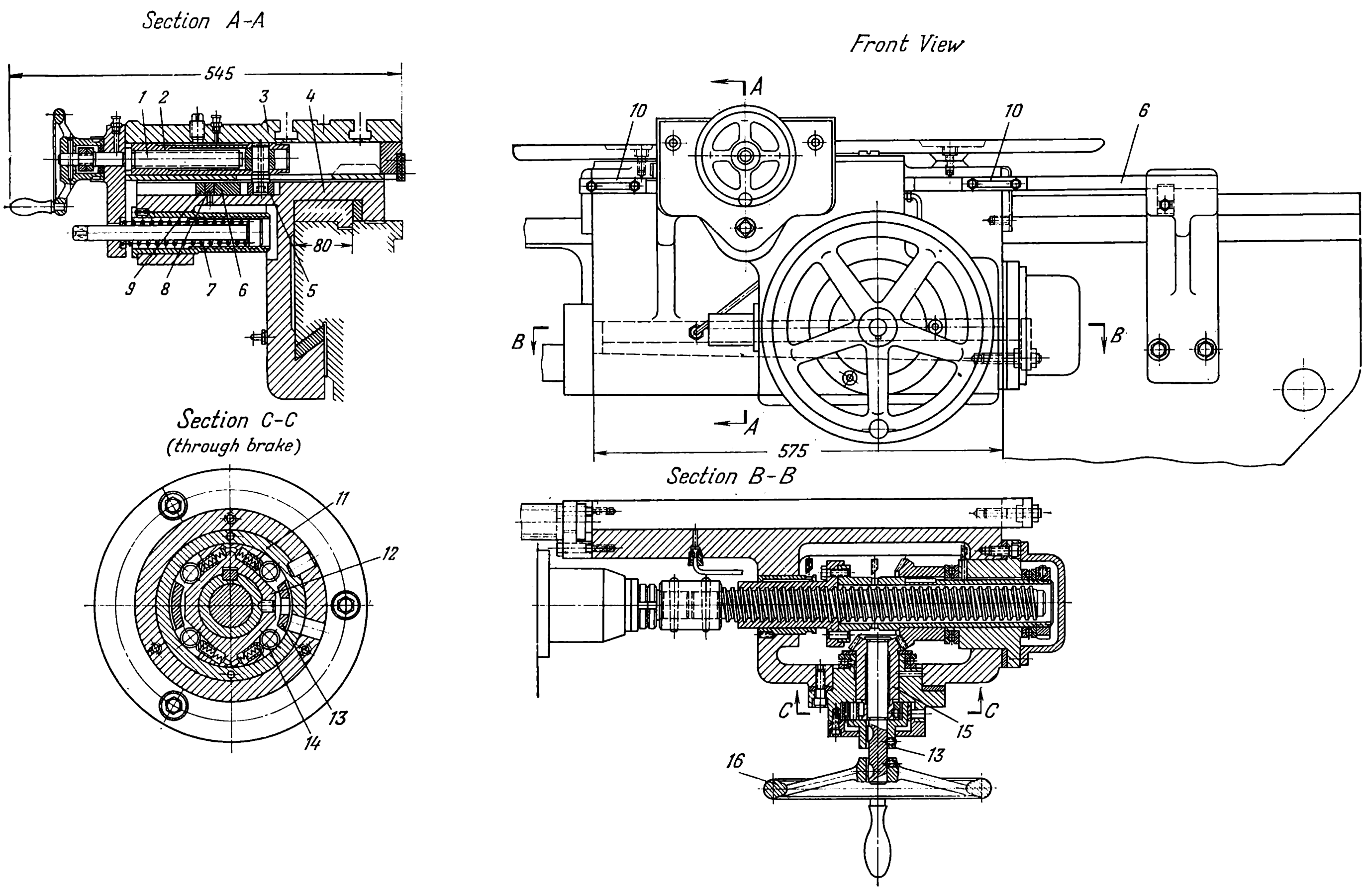


Fig. 52. Feed gearbox of the semiautomatic multiple-tool lathe, model 1730


Section A-A
545
1
2
3
4
5
6
7
8
9
80
Front View
A
A
10
10
6
B
B
575
Section C-C
(through brake)
11
12
13
14
Section B-B
C
C
15
13
16

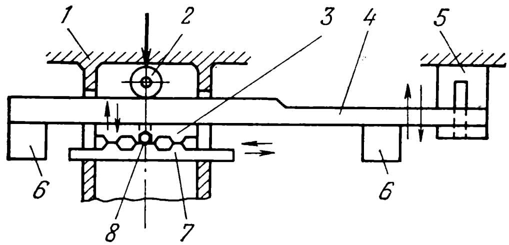


Fig. 54. Principle of the tool relief mechanism:

1—carriage saddle; 2—pin with a roller linked to the cross slide, which is forced back by a spring; 3—bar with bevelled recesses and projections, which is movable in the transverse direction; 4—cam bar, which slides in the transverse direction; 5—bracket with transverse guides for the slide block of the cam bar; 6—two stops of bar 7; 7—bar with bevelled recesses and projections, which is movable in the longitudinal direction; 8—pin holding bar 3 against longitudinal displacement

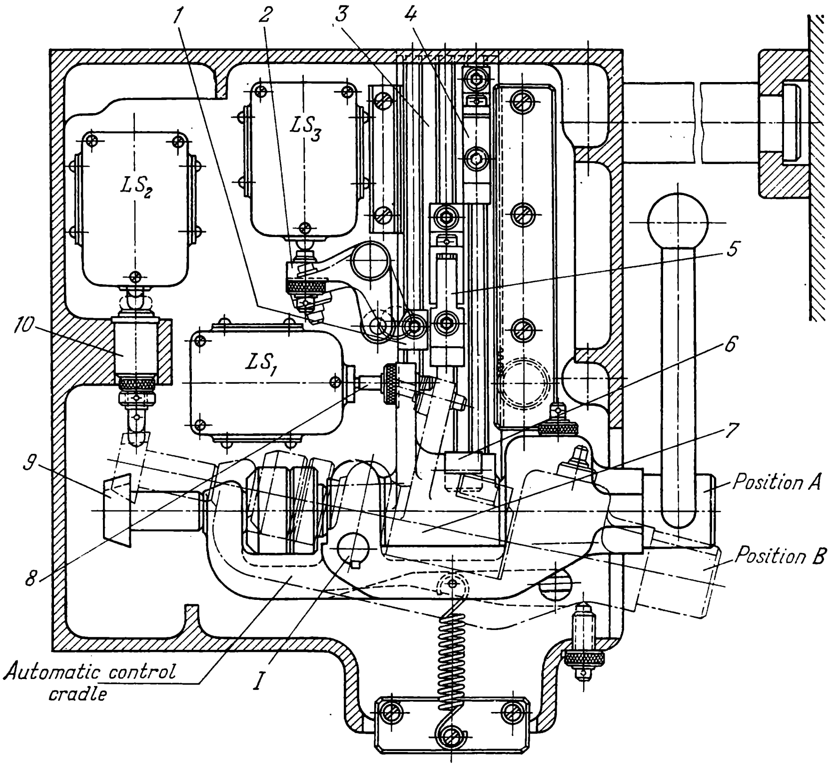


Fig. 55. Master control switch of the model 1730 multiple-tool semiautomatic

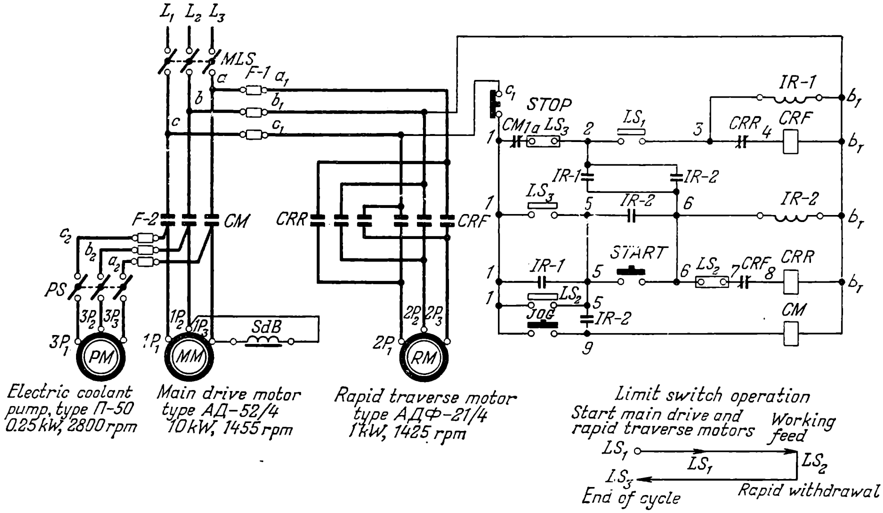


Fig. 56. Elementary electric circuit diagram of the model 1730 semiautomatic lathe

noid *SdB* which releases the band brake of the main drive. Contactor *CRF* is held closed by supply circuit *1* (*IR-1*), *5* (*IR-2*), *6* (*IR-1*), *2*, *3* and *4*.

As the cradle is raised, the jaw clutch of the working feed drive train is engaged; by means of the overrunning clutch, however, the rapid traverse motion is transmitted to the carriages from the rapid traverse motor.

During the approach movement, rack *3* (Fig. 55) with the trip dogs *1*, *4* and *5* travels downward. Trip dog *1* releases lever *2*, opening the contacts 1_a-*2* of limit switch LS_3.

Working feed of the carriages. In the descent of trip dog *5* (Fig. 55) its bevelled front end runs up against roll *6*, turning lever *7* with screw *8* about the cradle axis and thereby retracting screw *8* to the side from the pin of limit switch LS_1. This opens contacts *2-3* of LS_1 (Fig. 56), breaking the supply circuit of the coils of *CRF* and *IR-1* and switching off the rapid traverse motor *RM*. As the motor stops, working feed begins. The coil of contactor *CM* is supplied along the circuit *1* (LS_2 and LS_3 in parallel), *5* (*IR-2*), *9* and *1* (LS_3), *5*, *5*, *5* (*IR-2*) and *9*.

Rapid withdrawal. At the end of the working travel, trip dog *4* (Fig. 55) with its bevelled end runs against roll *6*, turning lever *7* farther about the cradle axis and retracting the locking member from its seat (Fig. 52). At this the cradle drops, disengaging the working feed drive clutch and then actuating pin *10* (Fig. 55) with trip dog *9*. Pin *10* depresses the pin of limit switch LS_2, closing contacts *6-7* (Fig. 56) which close contactor *CRR*. This contactor

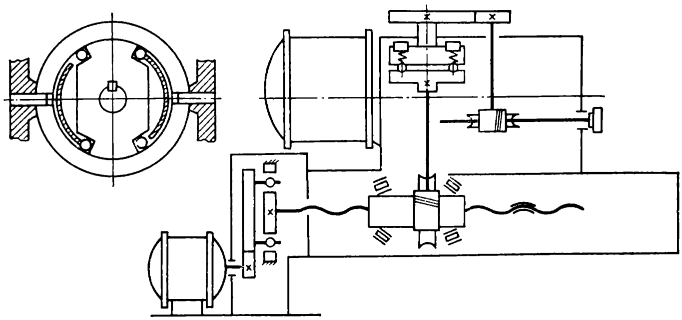


Fig. 57. Diagram of a power unit in which the lead screw nut rotates to provide working feed and the lead screw rotates for rapid traverse movements

switches on the rapid traverse motor for reverse rotation, returning the carriages rapidly to the initial position.

At the end of the withdrawal movement, trip dog *1*, travelling upward with rack *3* (Fig. 55), turns lever *2* which presses the pin of limit switch LS_3. This breaks the coil supply circuit of intermediate relay *IR-2* and thereby cuts off the supply of the coils in contactors *CM* and *CRR*. As a result, the main drive motor, rapid traverse motor and coolant pump motor are switched off, and the band brake solenoid is de-energized. The lathe stops.

Kinematic schemes with rotation of both lead screw and nut are used in power units (Fig. 57). Working feed is obtained with nut rotation driven from a shaft of the spindle head through a ball-type safety clutch which begins to slip when the unit runs up against a positive stop.

Rapid approach and withdrawal of the power unit along its saddle (guide plate) are accomplished by the lead screw which is driven by the rapid traverse motor through a two-direction reversible overrunning clutch whose cam member is keyed on the lead screw. Upon rotation of the lead screw, its nut is braked by the worm gearing. Upon nut rotation, two rollers of the overrunning clutch become wedged between the cam member and the stationary external housing and thus brake the screw.

The power unit operates on a semiautomatic cycle. It is controlled by trip dogs clamped on the unit and actuating limit switches mounted on the saddle.

To reduce overtravel of the unit in rapid traverse movements, the rapid traverse motor is furnished with an electric brake or with a time-delay relay if plugging is used to stop the motor.

The rapid traverse and working feed trains are joined by means of clutches and the screw-and-nut pair itself (Fig. 57). The overrunning clutch, as a rule, is of the reversible type (in contrast to the overrunning clutch of cam-controlled automatic and semiautomatic machine tools). When the rapid traverse

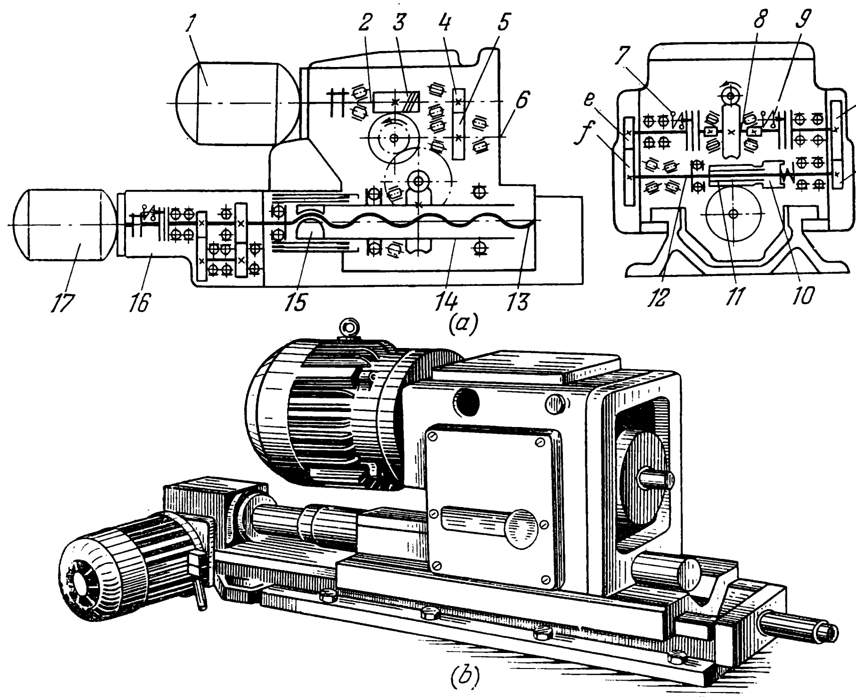


Fig. 58. Power unit manufactured by the Minsk Transfer Machine Plant:
(*a*) kinematic diagram; (*b*) general view

train is engaged and reversed, considerable acceleration and high inertia forces are developed because of the constant pitch of the lead screw, if the mass of the table and workpieces is great (semiautomatic planer-type milling machines). In such cases, epicyclic mechanisms are preferred for joining the rapid traverse and feed trains since they are more capable of carrying high inertia torques than an overrunning clutch.

The Minsk Transfer Machine Plant produces self-contained power units with lead screw feed (Fig. 58). From main drive motor *1* through shaft *2* and gears *4* and *5*, rotation is transmitted to shaft *6* which is linked to the spindle head. From shaft *2* rotation is transmitted through worm gearing *3* to shaft *8* on which electromagnetic clutches *7* and *9* are mounted. Upon engaging clutch *7* the working feed motion is transmitted to shaft *12* through change gears *e* and *f*. When clutch *9* is engaged rotation is transmitted through change gears *c* and *d* instead. Through safety clutch *10* and worm gearing *11*, shaft *12* drives sleeve *14* in which nut *15* of lead screw *13* is secured.

At the working feed, set up by installing the required change gears *e*, *f*, *c* and *d*, and switched over during the cycle by engaging either clutch *7* or *9*, the lead screw is held stationary by electromagnetic brake *16*. This brake is also used in disengaging rapid traverse of the power unit. Rapid approach and withdrawal of the unit are effected by reversible electric motor *17* which rotates lead screw *13* while nut *15* and sleeve *14* are held stationary by the self-locking worm gearing *11*.

Operation to a positive stop. The conditions of operation in travelling up to a positive stop, when the carriage is traversed by a lead screw, are considerably more complex than when a cam drive is used. In accordance with its profile, the cam provides reduction in the rate of feed, dwell of the required duration for facing operations, reversal and withdrawal of the carriage. All of these elements of the cycle can be obtained with a lead screw only by suitably designing the screw drive and its control system. To obtain precise dimensions in machining with such systems, the force exerted by the carriage against the positive stop and the consequent deflections in the carriage should be constant, provision being made for independent adjustment of this force. This is of prime importance for precise machining of diameters by tools in cross slides. If a cam drive is used, this is achieved by adjusting the stop, length of travel of the slide and the setting of the cutting tool. With a lead screw it is impossible to vary the force exerted by the carriage against the positive stop by adjusting the stop and length of slide travel. The force exerted during dwell cannot be maintained constant in the various possible designs of this mechanism.

Slide dwell against a positive stop is accomplished with a lead screw drive by either of two methods: (1) by using safety clutches that slip during this dwell, and (2) by mounting the longitudinal bearings of the screw on springs. Ball-type safety clutches are used for this purpose. In the case of high torques, use is made of jaw clutches which have nonuniform spacing of the jaws to facilitate slipping and clicking once each revolution. In a setup with dwell against a positive stop, the rapid traverse motor is switched over at the end of the working feed motion by a limit switch through a time-delay relay to rapid withdrawal. The advantage of this method is the constant load on the screw drive while the slide dwells against the positive stop. If the influence of the intermediate links of the drive is not taken into account, it may be assumed that this method ensures a constant force on the stop, and this is of prime importance insofar as machining accuracy is concerned.

The range of variation of the exerted force is limited by the operating conditions of the clutch as a safety member. The minimum force cannot be less than the maximum possible feeding force in the cutting process. This drawback, however, is not decisive and the use of slipping safety clutches is the principal method of obtaining slide dwell against a positive stop with a lead screw drive of the slide (see Figs. 52, 57 and 58).

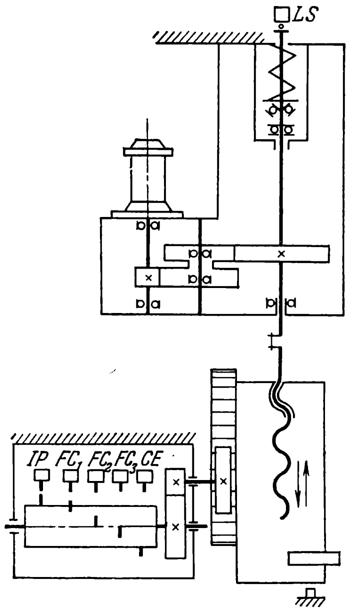


Fig. 59. Diagram of the longitudina carriage drive in a semiautomatic lathe: *IP*—initial position; *CE*—end of the cycle; FC_1, FC_2 and FC_3—feed changes

Another method of effecting dwell of a slide on a positive stop is shown in the diagram of the drive of the longitudinal carriage of a semiautomatic lathe (Fig. 59). Here a d-c electric motor, supplied by a rotary amplifier (amplidyne), is employed for cyclic change-overs in the speeds of the working and idle travel. The cycle is controlled by a master switch.

The screw bearings bear against Belleville (disk) springs to allow operation against a positive stop and to prevent overloading. In dwell against the positive stop the lead screw shifts axially with the spring-loaded bearing, tripping a limit switch which switches off the electric motor through a time-delay relay.

This method has the following disadvantages: (1) lack of smooth working feed due to the spring-loaded bearing of the lead screw, (2) variable force exerted by the carriage against the positive stop, increasing in the course of dwell and affecting the machining accuracy, and (3) excessive overloads in the mechanism during carriage dwell which may lead to breakage in setting-up.

These drawbacks indicate that slipping safety clutches are to be preferred for operations incorporating dwell.

CHAPTER 4

HYDRAULICALLY OPERATED AND CONTROLLED AUTOMATIC MACHINE TOOLS

4-1. General Properties of a Hydraulic Drive for Feeds and Auxiliary Motions

Hydraulic (fluid-power) drives are extensively applied in automatic and semiautomatic machine tools for transmitting motion to the main and auxiliary working members. This is due to the many advantages offered by fluid power systems (see also Part 4, Vol. 2), which include the following:

1. They permit large forces to be transmitted by hydraulic cylinders of comparatively small overall size.
2. They offer the means of obtaining infinitely variable rates of feed, thereby making available the most expedient cutting conditions and idle travel speeds, and also permit cyclic re-engagements of the working feeds.
3. They ensure smooth reversal and braking. A working member can be stopped at a preset moment; dwell operation to a positive stop is convenient.
4. They offer wide opportunities for the application of hydraulic safety devices and for interlocking the motions of various members.
5. The remote-control features of fluid-power systems enable spatial linkages between the drive elements of various working members to be readily and easily accomplished and make it a simple matter to transmit considerable amounts of energy along complex paths. This simplifies the design of operative mechanisms for working members.
6. A hydraulically controlled machine tool can be comparatively quickly changed over for machining other workpieces. This is of great importance in lot production when parts are machined in medium and small lots.
7. Hydraulic drives can be designed of standard elements which can be purchased from specialized plants.
8. Hydraulic drives do not, as a rule, have parts subject to high contact stresses. This eliminates many "bottlenecks" in production due to rejections of parts because of defects in material, heat treatment or grinding finish. Such "bottlenecks" are characteristic of the debugging period in mastering the production of new cam-controlled automatic and semiautomatic machine tools.

Along with the advantages, fluid-power drives have the following significant disadvantages which make it difficult to apply them in some cases:

1. The kinematic characteristic of the drive is not very rigid, especially at low speeds of displacement of the working members and low rates of oil

flow in the pumps and control valves. This drawback makes it difficult to obtain high accuracy and high-quality surfaces in finish machining.

2. The drive does not operate sufficiently smoothly in such cross turning operations as cutting grooves, form turning and turning with a broad-nose tool (turning the pins of crankshafts).

The effects of the two preceding disadvantages can be reduced (or even eliminated) by employing a hydromechanical drive in which motion is transmitted to the tool slide through a plate cam, cam bar (angular cam, see Fig. 115), or a lead screw driven by a hydraulic motor.

3. Except for the setting-up operation, the servicing of hydraulically controlled automatic machine tools is more involved than that of the cam-controlled type.

4. A complex control system of the fluid-power drive is required to obtain an automatic cycle in hydraulically controlled machines.

4-2. Automatic Cycle Control System

A power cylinder is a noncyclic operative mechanism. In this respect it is similar to a lead screw. Hence, the whole particular cycle of motions of a working member, driven by a hydraulic cylinder, is executed by its individual control system, operating by the principle of in-travel control of the given working member (see Sec. 3-2 and Fig. 46). The general cyclic interlinkage between the particular cycles of the various working members is accomplished, similar to an arrangement employing lead screws, by a hydraulic or electric control system. This system engages or switches on the cycle of motions of the given working member at a definite stage in the travel of another working member with which the first member is interlinked by a sequence of actions or by mutual interlocking (see Sec. 3-2 and Fig. 47). The cycle of motions of the given working member is switched off by the signal from its individual control system.

Thus, if there are several working members, a system of individual in-travel controls with mutual linkages and interlocking is employed.

Since a hydraulic power cylinder is a noncyclic operative mechanism, the functions of the control system are greatly extended in comparison with cam-controlled automatic machine tools, and the control system becomes more complicated.

At the same time, such an in-travel control system simplifies changing over the machine to a different workpiece, enabling the machine tool to be efficiently used in lot production.

The use of a hydraulic drive makes it possible to transmit command pulses and to actuate the control system not only as a function of the path travelled but also as a function of the pressure (travel up to a positive stop, safety devices, interlocking, etc.).

Hydraulic servosystems (with feedback) can be used to copy the given profile of a part and to reproduce motions specified by the command mechanism.

The command circuit of the control system may be either hydraulic, in which case we have a *hydraulic control system* or it may be electric with a hydraulic operative circuit of control. In the second case, we have an *electrical control system*.

In a hydraulic control system, command trip dogs and stops, linked kinematically with the travel of the working member, directly actuate the control pilot valves (piston or rotary type) and, through them, the main directional valves which control liquid flow to the power cylinders. In some cases, trip dogs or stops may control the directional valves directly.

In an electrohydraulic control system, electric pickups (limit switches, pressure switches, etc.), actuated by command trip dogs and stops, or by an increase in pressure in the hydraulic cylinder when the carriage runs up against a positive stop, transmit command signals through the electric control circuit to the corresponding solenoids. At low rates of oil flow in the power cylinders, the main directional valves are actuated by the solenoids themselves. In cases of high flow rates, the solenoids actuate pilot valves which, in turn, control oil flow to operate the main directional valves.

When a machine tool is built into an automatic transfer machine or line, the use of an electric command control circuit facilitates the centralization of controls at a main control desk. On the other hand, a large amount of electric contacting equipment makes an electrohydraulic control system less dependable than a purely hydraulic control system.

If a machine tool with a hydraulic control system is built into an automatic transfer machine or line, its automatic cycle is started by the general electric command control circuit.

The hydraulic elements of automatic cycle control systems of various working members are combined in control panels (see Part Four, Vol. 2), reducing the overall size of the system, facilitating standardization and simplifying assembly of the hydraulic equipment.

4-3. Power Unit with a Hydraulic System of Automatic Cycle Control

Hydraulic power units are available for light and heavy duty with an electric motor power range from 1 to 30 kW and an axial (feed) force from 500 to 11,000 kgf. The absence of parts subject to high contact stresses, good conditions of lubrication of parts subject to load and the dependable protection against overloads contribute substantially to the long service life of hydraulic power units. They are intended for drilling, boring and milling operations performed with simple linear cycles of motion: rapid approach

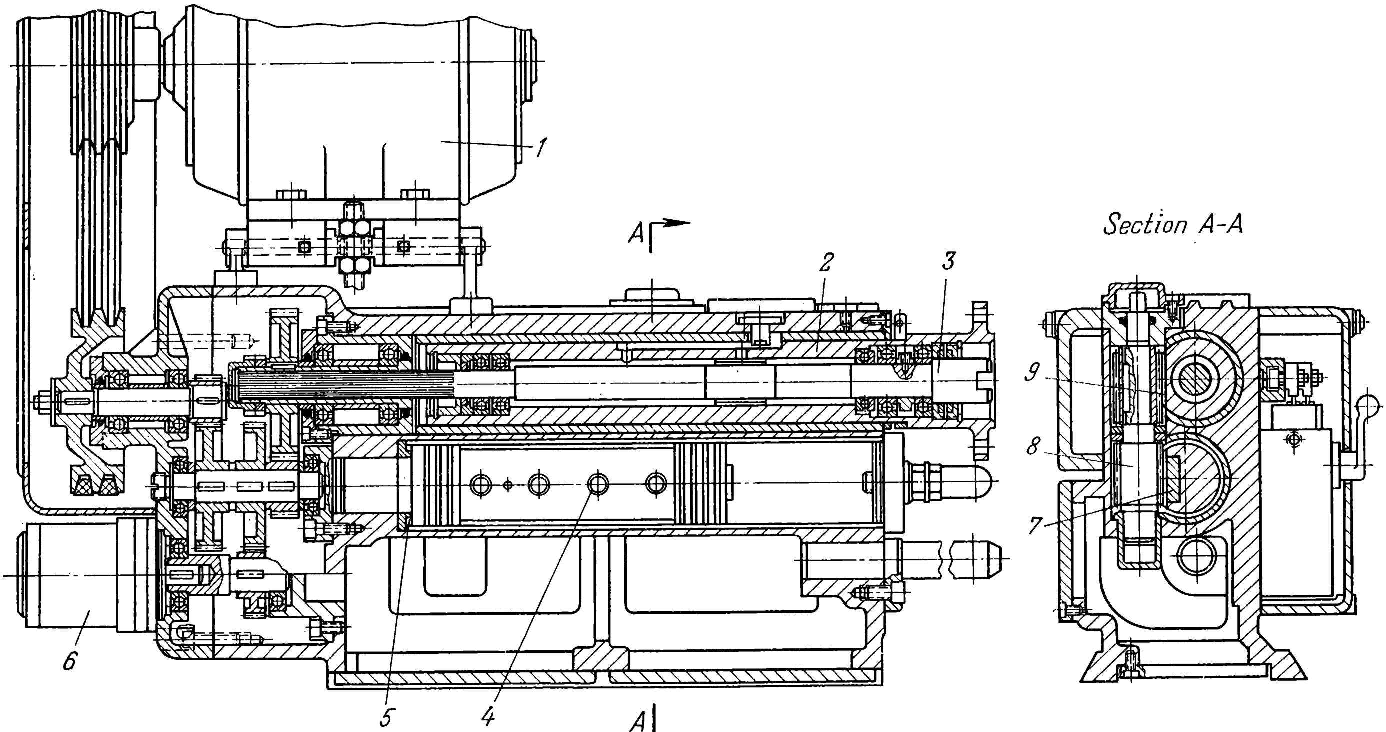


Fig. 60. ZIL Plant power unit

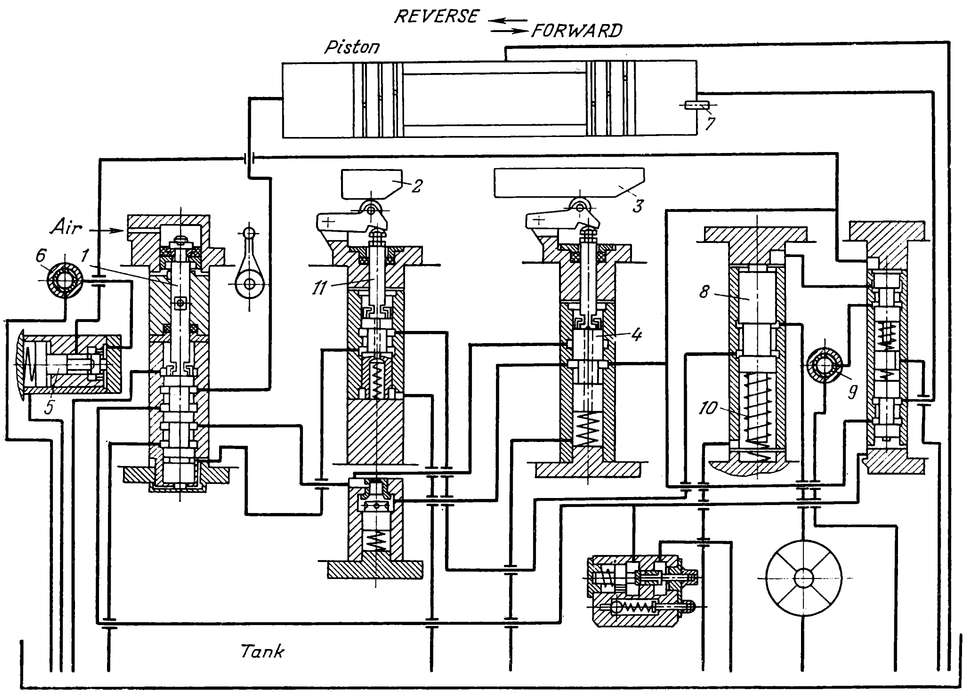


Fig. 61. Hydraulic circuit diagram of the ZIL Plant power unit

of the cutting tools; one or two working feeds; dwell against a positive stop; rapid return of the tools to the initial position and stop.

In the ZIL Plant power unit (Fig. 60), fixed-displacement pump *6* is driven by electric motor *1* which also powers spindle *3*. The pump delivers oil into cylinder *5* to actuate piston *4*. The piston carries rack *7* and transmits motion to spindle quill *2* through rack pinions *8* and *9* and the quill rack.

The power unit is started by a solenoid-controlled pneumatic valve which admits compressed air into the upper end of valve *1* (Fig. 61). Valve *1* admits the full stream of oil delivered by the pump into the pressure end of the cylinder to effect rapid approach of the cutting tools. From the other end of the cylinder the oil drains freely back to the tank. At the end of the rapid approach motion, working feed trip dog *3* actuates valve *4*. At this, oil from the right-hand end of the cylinder drains to the tank through reducing valve *5*, which maintains a constant pressure before the flow-control valve, and longitudinal-feed flow-control valve *6*.

At the end of the working travel, the piston runs up against positive stop *7* and the unit begins the dwell phase of the cycle. At this, the pressure in the drain end of the cylinder drops to zero after which time-delay relay *8* begins to operate. Flow-control valve *9* of this relay serves to regulate the time-delay value. When valve *8* of the relay, under the action of spring *10*, has forced all the oil contained in its chambers out through flow-control valve *9*, valve *11* is switched over, changing the oil flow to the other end of the cylinder and thereby returning the quill piston to its initial position.

Power units which do not have the dwell feature have no time-delay relay and the spindle quill is returned directly after valve *11* is actuated by the reverse trip dog *2*.

Tests conducted on this power unit show that the accuracy with which rapid approach is switched over to the working feed ranges from 0.5 to 1.5 mm while the accuracy of the dwell position in operation up to a positive stop depends upon the thermal deformations of the power unit housing. When the oil was heated to about 50°C, the dwell position of the tools was displaced 0.1 mm.

4-4. The Use of Hydraulic Servosystems for Automatic Cycle Control

Hydraulic servosystems (Fig. 62), used for carriage drives, consist of the following principal elements:

P—programme medium—the template (in tracing a profile) or the command mechanism that specifies the motion (in tracing motions);

Pu—pump delivering a stream of oil to the operative mechanism (hydraulic power cylinder) *O* of the slide working member *W*;

T—tracer, designed as a directional valve admitting oil to one or the other end of the cylinder in accordance with the direction the valve is displaced by the programme medium;

F—feedback rigidly linking the slide with the housing or, in some cases, with the spool of the tracer.

Because of the feedback, upon displacement of the tracer spool, the housing shuts off the inlet passage for the admission of oil to the power cylinder when

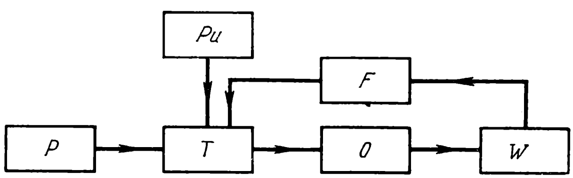


Fig. 62. Block diagram of a hydraulic servosystem

the slide has travelled a distance equal to the displacement of the tracer spool. If the spool moves continuously, a passage area of such magnitude is established that the slide travels at the same rate of speed as the valve spool. As a result, such a servosystem tends to maintain a constant distance between the stylus and the tool nose (in the direction of slide travel). This is accomplished, in contrast to a mechanical tracing system, with a relatively small pressure between the stylus and template.

As distinguished from electrical servosystems, hydraulic ones do not require amplification of the power of the signals and, due to their lower inertia, have much better quick-action features.

A drawback of hydraulic servosystems, one common to all control systems with feedback in general, is a tendency to instability when high accuracy and tracing speed are required.

4-5. Transverse Tracing with Automatic Variation of the Rate of Longitudinal Feed (Two-Dimensional Controlled Tracing)*

Stylus *3* of tracer *2* (Fig. 63), in contact with template *4*, shifts the valve spool in the direction which corresponds to the rise or fall of the template profile. This admits oil delivered by pump *5* to one end of cross slide cylinder *1* which travels in the direction determined by the rise or fall of the template profile. The oil forced out of the other end of the cylinder drains through flow-control valve *6* to the tank.

Since the housing of the tracer is rigidly linked to the cross slide, the slide will continue to travel until the passage between the valve spool and housing is closed by the motion of the slide (and valve housing). This shuts off oil flow to the corresponding end of cylinder *1*. Thus the slide reproduces the motion of the stylus tip.

During the tracing operation, longitudinal cylinder *9* traverses the saddle of the carriage from right to left. Oil from the left end of this cylinder is forced out through automatic regulator *8* and the longitudinal-feed flow-control valve *7* back to the tank.

The automatic regulator is a two-land valve. Flats on the larger land of the spool throttle the oil while a pressure equal to that of the oil draining from the transverse cylinder acts on the end face of the smaller land. Upon an increase in the rate of cross feed, the pressure before flow-control valve *6* increases, compressing the spring of valve *8* to shift the spool. This reduces the area of the passage between the flats of the spool and the valve body,

* The circuit and method of calculation were developed by B. Korobochkin, Cand. Sc. (Eng.).

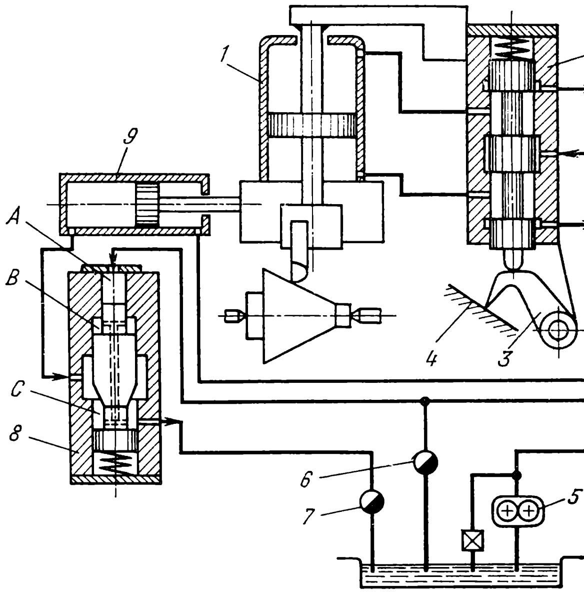


Fig. 63. Diagram of a hydraulic tracer-controlled servosystem with variation of the longitudinal feed rate

thereby increasing the throttling action. As a result, the rate of longitudinal feed is correspondingly reduced.

The law of resultant variation of feed along the workpiece profile can be derived as follows. Ignoring the friction forces and the reaction of the oil stream, the condition of static equilibrium of the regulator spool can be expressed by the equation

$$p_1 f_1 + p_2 f_2 = N_1 \tag{14}$$

where p_1 = pressure before the cross-feed flow-control valve and in chamber A of automatic regulator 8

p_2 = pressure before the longitudinal-feed flow-control valve and in chamber B of the automatic regulator which is connected to chamber C by channels in the valve spool

f_1 = area of the valve spool land in chamber A

f_2 = annular area of the valve spool land in chamber B

N_1 = constant force of the valve spring which has a flat characteristic.

Expressing the velocity of oil flow through the orifice of the cross-feed flow-control valve on the basis of the continuity equation and oil exit con-

ditions (see Part Four, Vol. 2), we obtain

$$\frac{s_1F_1}{\Delta_1}=k_1\sqrt{\frac{2g}{\gamma}p_1} \tag{15}$$

and, in the same way, for the oil flow rate in the orifice of the longitudinal-feed flow-control valve

$$\frac{s_2F_2}{\Delta_2}=k_2\sqrt{\frac{2g}{\gamma}p_2} \tag{16}$$

where s_1 = rate of cross feed of the tracing slide
s_2 = rate of longitudinal feed
k_1 and k_2 = dimensionless discharge coefficients in the orifices of the cross- and longitudinal-feed flow-control valves, respectively
Δ_1 and Δ_2 = areas of the orifices of the cross- and longitudinal-feed flow-control valves, respectively
γ = specific weight of the oil
F_1 = effective piston area of the cross-feed cylinder
F_2 = area at the exit end of the longitudinal-feed cylinder.

Substituting the values of p_1 and p_2 from equations (15) and (16) into equation (14) we can write

$$\frac{s_1^2}{\dfrac{2(\Delta_1k_1)^2gN_1}{F_1^2\gamma f_1}}+\frac{s_2^2}{\dfrac{2(\Delta_2k_2)^2gN_1}{F_2^2\gamma f_2}}=1 \tag{17}$$

which is the equation of an ellipse with the semiaxes

$$a=\sqrt{\frac{2gN_1(\Delta_1k_1)^2}{F_1^2\gamma f_1}}=\frac{\Delta_1k_1}{F_1}\sqrt{\frac{2gN_1}{\gamma f_1}} \tag{18}$$

$$b=\sqrt{\frac{2gN_1(\Delta_2k_2)^2}{F_2^2\gamma f_2}}=\frac{\Delta_2k_2}{F_2}\sqrt{\frac{2gN_1}{\gamma f_2}} \tag{19}$$

Since s_{pr}, the feed along the profile, is equal to the geometric sum of the longitudinal and cross feeds, i.e.,

$$s_{pr}^2=s_1^2+s_2^2 \tag{20}$$

equation (17) represents the law of the variation of feed along the workpiece profile for various angles of slope of the profile (see Fig. 63).

It can be seen from equation (17) that at definite constructional dimensions of the regulator, the ratio of the semiaxes of the ellipse depends only on the setting of the flow-control valves and the corresponding discharge coefficients of their orifices. If the regulator valve spool is designed to comply with the condition

$$\frac{f_1}{f_2}=\left(\frac{F_2}{F_1}\right)^2 \tag{21}$$

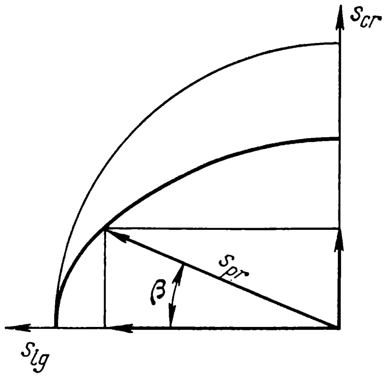


Fig. 64. Curves of feed variation along the workpiece profile

then the ratio of the semiaxes of the ellipse will be

$$\frac{a}{b} = \frac{\Delta_1 k_1}{\Delta_2 k_2} \tag{22}$$

At $\Delta_1 k_1 = \Delta_2 k_2$, the ratio $\frac{a}{b} = 1$, i.e., equation (17) is the equation of a circle. This conforms to the law of constant resultant feed along the profile of the workpiece.

The curves in Fig. 64 show the variation in the geometric sum of the feeds for cases of a constant geometric sum and for an elliptical law of resultant feed variation.

In the latter case, the feed along the profile has a definite value for each slope angle β of the profile. An elliptical feed variation law is used in turning stepped shafts when the cross feed should be less than the longitudinal feed.

The servosystem described can be used to turn cylindrical surfaces and end faces to a template in a single pass on lathes having the cross slide arranged square to the line of centres.

Contrary to the general method of increasing the output of an automatic lathe by combining the working travel motions and by multiple-tool machining, single-tool turning to a template requires much less time for setting up and readjustments and enables the minimum admissible design life of the cutting tool to be reduced from $4\frac{1}{2}$ hours to 45 min, correspondingly increasing the cutting speeds and feeds, and thereby increasing the output of the machine. Another factor contributing to the increase in productivity is the reduction in the number of passes and, therefore, in the time lost in idle motions of the slides and in handling motions.

In their production capacity, tracer-controlled semiautomatics successfully compete with the multiple-tool models.

Servosystems with automatic variation of the rate of longitudinal feed have been applied in a number of tracer-controlled multiple-tool semiautomatic lathes manufactured by the Orjonikidze Machine Tool Plant (models 1731C, 1722, 1712, etc.).

Variations in the cutting speed in tracer-controlled turning have a detrimental effect on the operation of chip breakers. Consequently, automatic cyclic variation of the spindle speed, to suit a varying workpiece diameter, and improvement of chip disposal conditions by a more expedient arrangement of the carriage unit are factors of significance for the production capacity of these lathes.

4-6. Tracer-Controlled Semiautomatic Lathe, Model 1712 (Orjonikidze Machine Tool Plant in Moscow)

This semiautomatic is intended for turning shafts having cylindrical, tapered or formed sections by means of a hydraulic tracer-controlled slide, and for facing and grooving by means of one or two cross slides.

The machine has a special mechanism (template drum) for changing the templates during the operation. This enables the tracer-controlled slide to operate in several passes, or cuts (up to four), in each automatic cycle. This feature is of especial advantage in machining blanks having large or nonuniform allowances, i.e., in turning stepped shafts from bar stock. The use of a two-speed electric motor in the main drive permits the spindle speed to be changed during the automatic cycle. This motor, change gears and two sliding double cluster gears provide 24 different spindle speeds. The kinematic diagram of the lathe is shown in Fig. 65.

The high rigidity of the lathe (resulting from the frame-type design) and the amply powered drive enable carbide-tipped tools to be used to their full capacity in cutting a heavy chip.

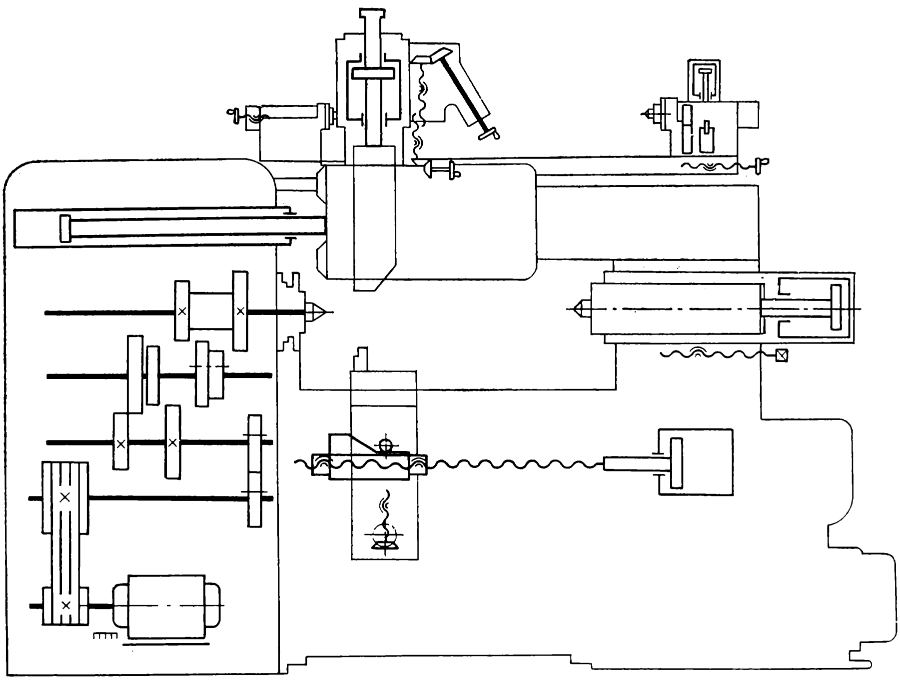


Fig. 65. Kinematic diagram of the tracer-controlled semiautomatic lathe, model 1712

Solenoid Operation Schedule for the Facing Slide Carriage

Operation elements	Solenoids Sd'_1	Sd'_2
Stop	—	—
Rapid approach	+	+
Working feed	+	—
Rapid withdrawal	—	+

Solenoid Operation Schedule for the Tracer-Controlled Slide Carriage

Operation elements	Sd_1	Sd_2	Sd_3	Sd_4	Sd_5	Sd_6
Stop	—	—	—	—	—	—
Rapid longitudinal approach	+	+	—	—	—	—
Rapid cross approach	—	—	—	+	+	—
Skip traverse in section of first working feed	+	+	—	+	+	—
Skip traverse in section of second working feed	+	+	+	+	+	—
Tracer-controlled turning at second working feed	+	—	—	—	+	—
Tracer-controlled turning at first working feed	+	—	+	—	+	—
Rapid withdrawal	—	+	—	+	—	—
Template drum indexing	—	—	—	—	—	+

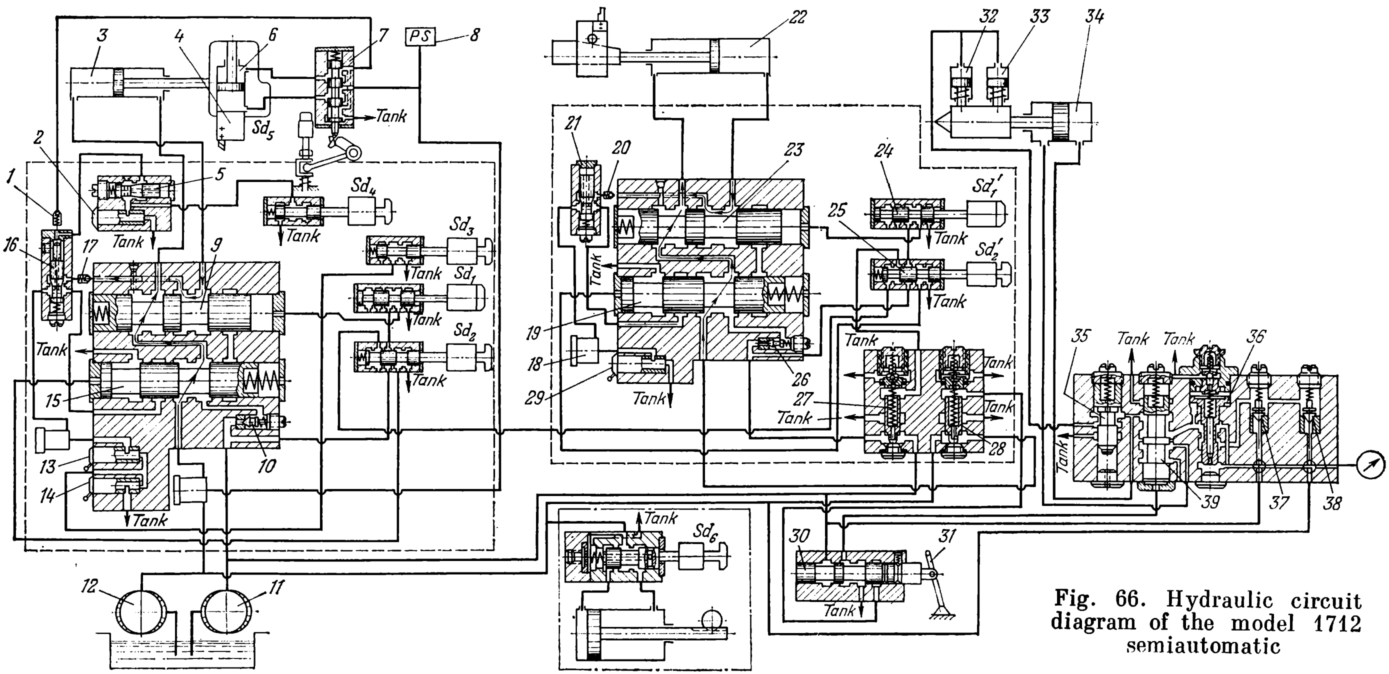


Fig. 66. Hydraulic circuit diagram of the model 1712 semiautomatic

Hydraulic Drive

Each of the carriages of the lathe has its own electrohydraulic control panel and can operate on an independent cycle. The interaction of the carriages during the automatic cycle is accomplished by a control system which provides the following combinations of carriage and slide operation:

(a) both carriages begin travel simultaneously;

(b) the tracer-controlled slide begins first and, at a preset point in any pass, the facing slide begins operation;

(c) the facing slide begins first and, after it completes its travel, the tracer-controlled slide is engaged.

The hydraulic control panels of the carriages (Fig. 66) are arranged so that oil can be admitted to each hydraulic cylinder either from the high-pressure pump with delivery from the low-pressure pump shut off, or from the low-pressure pump with delivery from the high-pressure pump shut off.

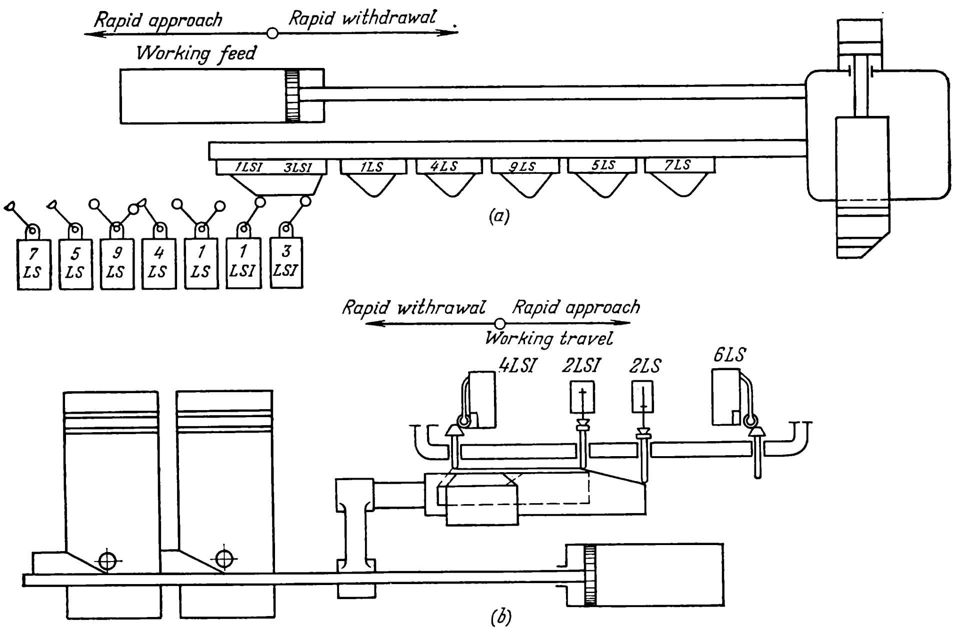


Fig. 67. Limit switches and trip dogs controlling slide and carriage motions:
(*a*) tracer-controlled slide carriage (*1LSI* and *3LSI*—trip dogs interlocking the electric circuit and carriage stop in the initial position; *1LS*—trip dog for rapid cross approach; *4LS*—trip dog for starting facing slide operation; *9LS* and *5LS*—trip dogs for engaging skip traverse and second rate of feed, respectively; *7LS*—trip dog for disengaging skip traverse); (*b*) facing slide carriage (*4LSI* and *2LSI*—trip dogs interlocking the electric circuit and carriage stop in the initial position; *6LS*—trip dog for reversing; *2LS*—trip dog for working feed)

The pressure in the system is maintained constant. The carriages can operate independently.

The operation of the hydraulic systems of the carriages is controlled by solenoids which receive command signals from limit switches and pressure switches. The required sequence, with which the various units are switched on or off in the operating cycle, is obtained by setting up trip dogs which operate limit switches (Fig. 67).

The pump complex consists of two pumps with a common shaft having the following deliveries: 50 litres per min at a pressure of 10 to 15 kgf per sq cm for the rapid traverse (low-pressure) pump and 12 litres per min at a pressure of 20 to 25 kgf per sq cm for the feed (high-pressure) pump. When both carriages are travelling at the rate of working feed or are stationary at the initial positions, the low-pressure pump is unloading to the tank. The high-pressure pump is unloaded to the tank upon retraction of the tailstock spindle.

The hydraulic control panel of the tracer-controlled slide provides the following cycle of operation: (a) rapid longitudinal approach of the carriage; (b) rapid cross approach of the tracing slide; (c) tracer-controlled turning at the first and second rates of working feed and rapid skip traverse over the parts of the workpiece not subject to machining; (d) rapid withdrawal and (e) initial position (STOP).

During each pass of the tracer-controlled slide, the speed and feed can take two preselected values. These can be alternated as many times as the number of trip dogs that have been set up. The spindle speed and the tool feed can be changed independently or concurrently (the feed is changed automatically upon a change in spindle speed).

The hydraulic control panel of the cross-feed carriage provides the following operating cycle: rapid approach, working feed, rapid withdrawal and stop in the initial position.

The hydraulic control panel of the tailstock provides for advancing, clamping and retracting the tailstock spindle.

Operation of the Hydraulic Control Panel of the Tracer-Controlled Slide

This panel is designed for electrohydraulic control of the automatic cycle and operates on the principle of a two-dimensional controlled tracing system (see Fig. 63).

STOP position. In the STOP position, the solenoids of all the pilot valves are de-energized. Therefore, directional valve *9* (see Fig. 66) is held by spring action in the extreme right-hand position (on the diagram) and directional valve *15* in the extreme left-hand position. At this, valve *9* shuts off flow from the high-pressure pump *12* to the end of cylinder *3* for longitudinal travel of the tracer-controlled slide carriage. Load-dividing valve *10* shuts off

oil flow from the low-pressure pump *11*. Tracer spool *7* shuts off oil flow to cross-feed cylinder *6* of the tracer-controlled slide.

Rapid longitudinal approach. Solenoids Sd_1 and Sd_2 are energized, the rest remain de-energized. At this, valve *9* is shifted by oil pressure to the left-hand position and valve *15* is shifted to the right. Oil from the high-pressure pump is not admitted to the panel because valve *15* closes off the corresponding recess in the panel. Oil is admitted to the right end of longitudinal-feed cylinder *3* from pump *11* through dividing valve *10* (serving also as a throttling device for rapid traverse motions) and grooves in the directional valves. From the left end of the cylinder oil is drained to the tank through grooves in the directional valves, check valve *17* and automatic regulator *16*, by-passing the working feed flow-control valves.

Rapid cross approach. The command pulse for rapid cross approach of the tracing slide to the blank is transmitted by a limit switch. This de-energizes solenoids Sd_1 and Sd_2, and energizes solenoids Sd_4 and Sd_5. Under the action of their springs the directional valves are shifted back to their initial positions (as in the STOP position). Therefore, oil flow to and from the longitudinal-feed cylinder is shut off.

Solenoid Sd_5 releases tracer spool *7* which shifts to the lower position, admitting oil to the lower end of cross-feed cylinder *6*. Solenoid Sd_4 shifts the pilot valve to its left-hand position in which oil drains from the upper end of cylinder *6* through check valve *1*, reducing valve *5* and back to the tank, by-passing cross-feed flow-control valve *2*. Consequently, the tracing slide moves downward rapidly until the stylus lever contacts the template. When this occurs, the stylus valve occupies its neutral position in respect to the valve housing. At this, pressure switch *8*, previously connected to the tank and subject only to the atmospheric pressure, is connected to the high-pressure line. The closing of the pressure switch transmits a command pulse to switch over the solenoids to obtain tracer-controlled turning at the first working feed.

Tracer-controlled turning at the first working feed. Solenoids Sd_1 and Sd_5 are energized, the rest are de-energized. Directional valve *15* remains in the left-hand position, while directional valve *9* is shifted to the left-hand position by the pressure of oil admitted through the pilot valve of solenoid Sd_1. At this, oil is delivered by pump *12* through the grooves of the directional valves to the right end of the longitudinal-feed cylinder, while from the left end the oil flows through the grooves of directional valve *9* to automatic regulator *16* and longitudinal-feed flow-control valve *13*. From here it drains through the pilot valve of solenoid Sd_3 back to the tank. Since solenoid Sd_5 has freed the stylus tip, oil from pump *12* is admitted to the tracer spool *7* which controls the position of tracing slide *4* in the transverse direction. The discharge cavity of the stylus valve is connected to automatic regulator *16* and cross-feed flow-control valve *2* (oil cannot pass through the valve of sole-

noid Sd_4 because the solenoid is de-energized). The automatic regulator co-ordinates the longitudinal feed of the carriage with the cross feed of the tracing slide.

Tracer-controlled turning at the second working feed is accomplished similar to that at the first rate of feed except that solenoid Sd_3 is also energized. At this, oil from the first longitudinal-feed flow-control valve *13* passes to the second longitudinal-feed flow-control valve *14*. Oil drain through the pilot valve of solenoid Sd_3 is shut off since the solenoid is energized and holds the valve in its left-hand position.

Rapid withdrawal of the longitudinal carriage. The command pulse for rapid withdrawal is transmitted by a limit switch which is operated at the end of the working travel of the carriage. At this, solenoids Sd_2 and Sd_4 are energized while the rest are de-energized. Directional valves *9* and *15* are in their right-hand positions. Oil delivered by high-pressure pump *12* cannot enter the longitudinal-feed cylinder since it is shut off by directional valve *15*. From low-pressure pump *11*, oil is delivered through the directional valve and the grooves of the other directional valves to the left end of the longitudinal-feed cylinder while oil from the right end drains back to the tank. At the same time, since solenoid Sd_5 is de-energized, tracer spool *7* is shifted to the upper position by the spring with the aid of the lever. At this, oil is admitted through the stylus valve to the upper end of the cross-feed cylinder while from the lower end oil drains back to the tank, by-passing the cross-feed flow-control valve (since solenoid Sd_4 is energized). Thus, the carriage rapidly returns to its initial position. Close to this position a trip dog operates a limit switch which puts the control panel into the STOP position.

Hydraulic Control Panel of the Facing Slide Carriage

Rapid approach. The command pulse for rapid approach is given by the energizing of solenoids Sd_1' and Sd_2'. This shifts pilot valves *24* and *25* to their left-hand positions (on the diagram), admitting oil from rapid traverse (low-pressure) pump *11* to the end chambers of the directional valves. Then the spool of directional valve *19* is shifted to the right and that of valve *23* to the left. Oil delivered by the low-pressure pump passes through low-pressure valve *27*, dividing valve *26* and the grooves of the directional valves to the rod end of cross-slide cylinder *22*. From the right-hand (head) end of this cylinder, oil passes through a groove of directional valve *23*, check valve *20*, reducing valve *21* and a groove of valve *19* to the tank, by-passing the working-feed flow-control valve *29*.

Oil from high-pressure pump *12* cannot enter the panel since the spool of directional valve *19* closes off the corresponding recess in the panel.

Working feed. The command pulse for the working feed is obtained from a limit switch which de-energizes solenoid Sd_2'; solenoid Sd_1' remains energized. The spool of pilot valve *24* is shifted by the spring to the right. This connects the left end of valve *19* to the tank and the spring shifts valve *19* to the left. At the working feed, the pressure in the system increases and dividing valve *26* separates the high-pressure circuit from the low-pressure circuit. Oil is delivered by pump *12* through high-pressure valve *28* and the grooves of the directional valves to the left end of the cylinder. From the right end, oil drains through the groove of directional valve *23*, check valve *20*, reducing valve *21*, filter *18* and the working-feed flow-control valve *29* to the tank.

The piston of the cylinder travels at a velocity set up by means of flow-control valve *29*.

Rapid withdrawal. Rapid withdrawal is accomplished after a command pulse is received from a limit switch. This de-energizes solenoid Sd_1' and energizes solenoid Sd_2'. The spools of the directional valves are shifted (*23* by spring action and *19* by oil pressure) to the positions required for rapid withdrawal. Oil is delivered by pump *11* through the grooves of the directional valves to the right end of the cylinder. From the left end oil drains through the grooves of the directional valves back to the tank, by-passing flow-control valve *29*.

As in the case of rapid approach, oil from pump *12* does not enter the control panel.

Rapid withdrawal continues until the working member returns to its initial position in which it holds down the pin of the INITIAL POSITION limit switch.

Initial position (STOP). The limit switch transmits a command pulse for de-energizing solenoid Sd_2'. The directional valves are shifted to the STOP position, in which they shut off oil delivery from both pumps to the control panel

Hydraulic Drive of the Tailstock Spindle

Advancing and clamping the tailstock spindle. After the blank is fed out to the line of centres, the operator presses lever *31* of the pilot valve for tailstock spindle control, shifting valve *30* to the clamping position. In this position, the end chamber of reversing valve *39* of the spindle is connected to the tank so that the valve spool is shifted by spring action to the advance position. Opening check valves *37* and *38*, oil from both pumps passes through reducing valve *36* and the groove in the reversing valve to the right end of advancing cylinder *34*. From the left end oil drains through reversing valve *39* to the tank. This accomplishes rapid advance of the tailstock spindle until the blank is held up against the headstock centre. After this, the pressure in the system increases and oil, delivered only from high-pressure pump *12*, passes to the

shoulder of the spool of spindle clamping valve *35*, shifting this valve to the CLAMP position shown in the diagram. At this, clamping cylinders *32* and *33* are connected to the oil under pressure and the tailstock spindle is clamped.

The maximum clamping pressure is determined by the setting of reducing valve *36* of the control panel.

Retracting the tailstock spindle. After shifting pilot valve *30* to the UNCLAMP position, oil under pressure is admitted to the end chamber of reversing valve *39*, shifting the spool upward (to the RETRACT position) and compressing its spring. At this, the recess under the shoulder of the spool in clamping valve *35* is connected with the tank; the valve spool shifts downward by the action of the spring. This connects the upper ends of clamping cylinders *32* and *33* with the tank. Under the action of their springs the rods of the clamping cylinders are forced upward, thereby releasing the tailstock spindle.

Oil delivered by both pumps passes through check valves, the reducing valve and grooves of the reversing valve to the left end of the spindle cylinder. As a result, the tailstock spindle is rapidly retracted to the initial position.

Tailstock spindle in the initial position. Pilot valve *30* of the spindle is in the UNCLAMP position (the workpiece is removed from the lathe). The carriages and slides are in their initial positions, therefore the directional valves of the tracing and facing slides are in the STOP position. Low-pressure pump *11* is unloaded by valve *27* to the tank. The unload pressure of this pump holds reversing valve *39* in the RETRACT position.

Hydraulic tracer (Fig. 68). This unit is one of the most important in the lathe; its operation determines the machining accuracy. Tracers manufactured in the Orjonikidze Plant ensure precise operation of the system and a small inert zone. This is achieved by the provision of practically zero lap between the lands of the spool and the port recesses of the tracer body, enabling the leakage in the tracer to be held at a minimum.

In most tracing valves with zero lap, the spool is guided only in the end guide bushings. The orifices between the spool and the bushings are annular and this leads to the rounding of the edges of the lands and bushings in the process of continuous spool motion about the neutral position and, consequently, to an increase of leakage in the valve. In the tracing valve described above, the orifices are formed by longitudinal rectangular slots in the bushings. Thanks to this feature, the spool has guidance over its full length. This, in turn, keeps the edges of both the lands and bushings sharp, ensuring minimum leakage in the valve.

Template indexing mechanism. This mechanism (Figs. 69 and 70) enables the tracer-controlled slide to operate in several passes (up to four) in an automatic cycle. The mechanism consists of: (1) template drum *11* mounted

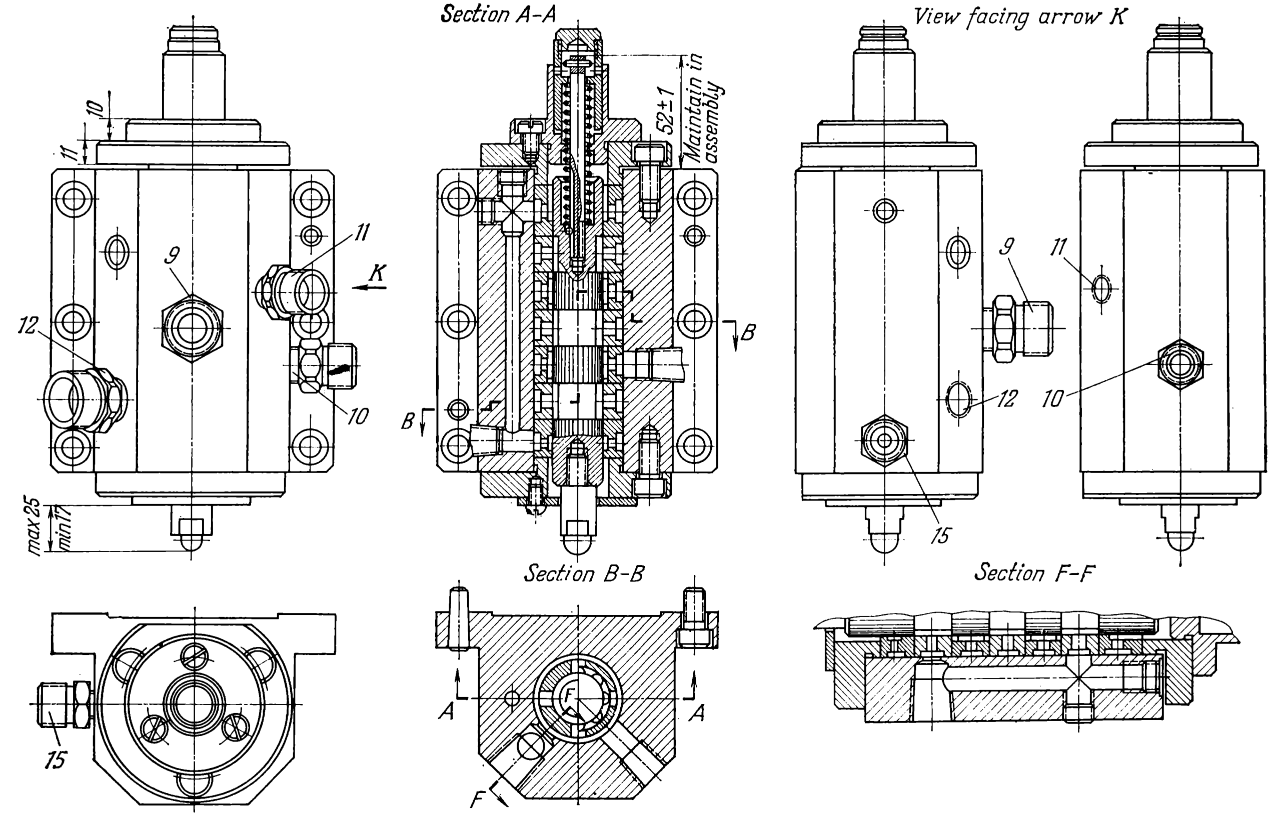


Fig. 68. Hydraulic tracer of the model 1712 semiautomatic lathe:

Connections: *9*—with the hydraulic control panel of the tracer-controlled slide carriage; *10*—with the lower end of the tracer-controlled cylinder; *11*—with the upper end of the tracer-controlled cylinder; *12*—with the hydraulic control panel of the tracer-controlled slide carriage; *15*—with the tank (see Fig. 66)

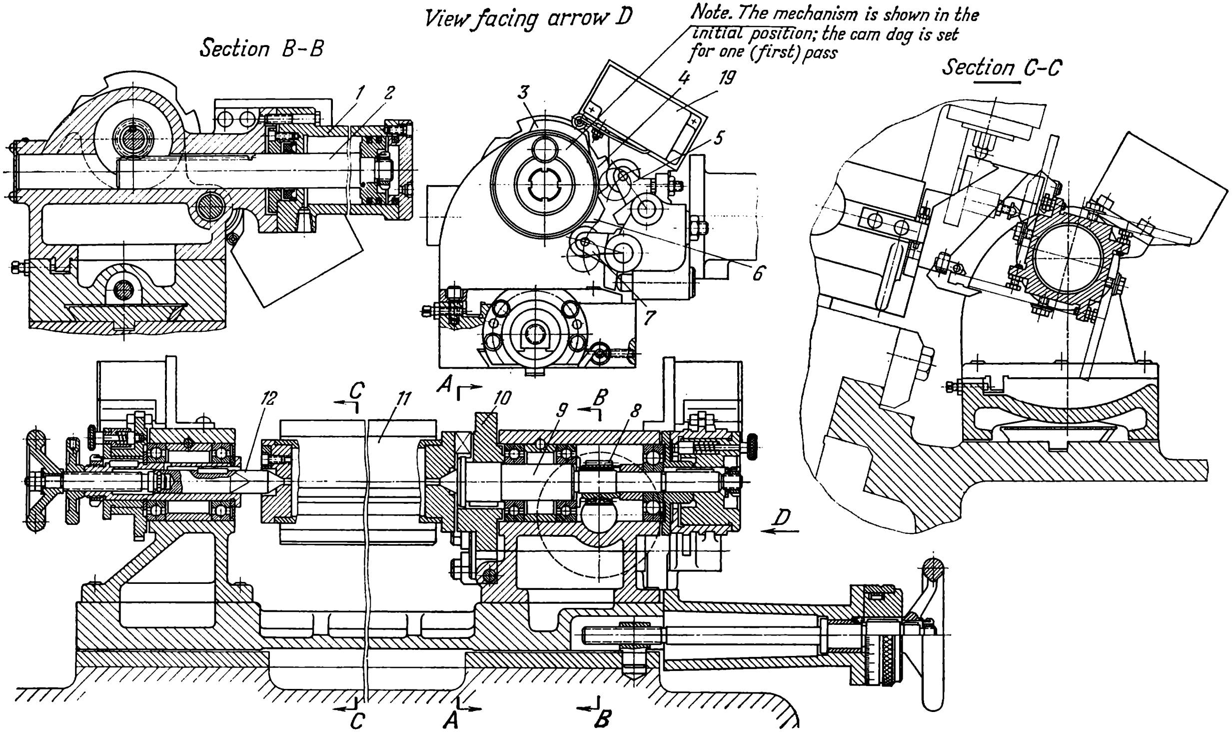


Fig. 69. Template indexing mechanism

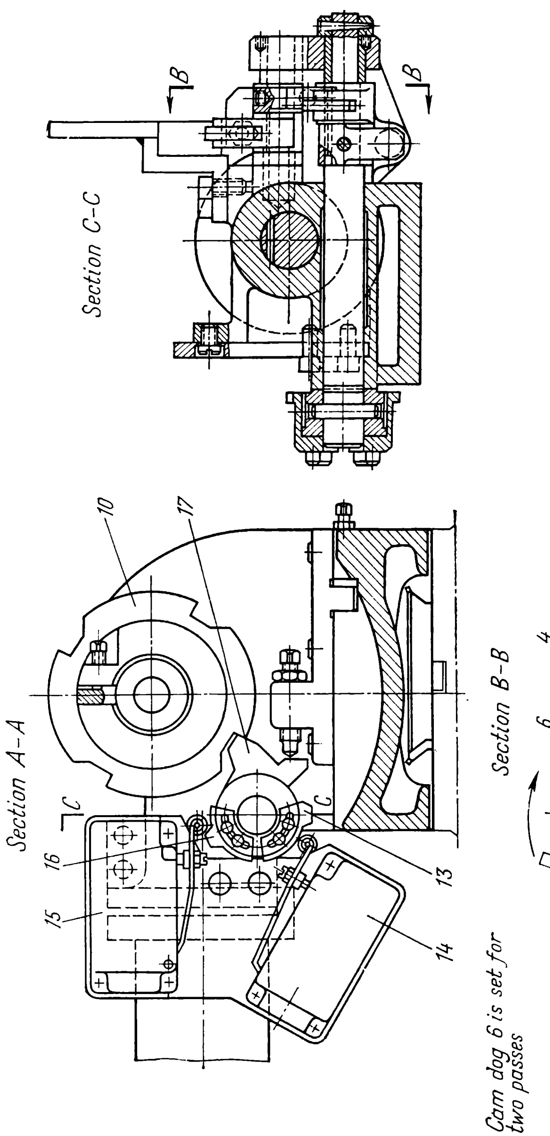


Fig. 70. Template indexing mechanism

between rotating centres *9* and *12*; (2) hydraulic cylinder *1* for indexing and locking the template drum; and (3) levers and trip dogs for setting up and locking the mechanism. This mechanism operates as follows.

After completing the first pass, the tracer-controlled slide returns rapidly to the initial position. In doing so it operates a limit switch which transmits a command pulse to energize solenoid Sd_6 (see Fig. 66). Oil passes through the control valve to the right end of cylinder *1* for indexing and locking the template drum. Under the action of the entering oil, piston *2* begins to move to the left, rotating centre *9* by means of rack pinion *8*. The template drum *11* rotates together with the centre. At this, the bevel of index plate *10* forces lever *17* out of its slot and cam *16* operates the limit switch *15*. The rotation of template drum *11* continues until lever *17* drops into the next slot of the index plate *10*. The pin of limit switch *15* is released and it transmits a command pulse for reversing piston motion. Centre *9* and template drum *11* rotate in the reverse direction until the side of the slot in the index plate runs against the edge of lever *17*. This locks the template drum in a new position.

Consecutive indexing and locking of drum *11* continue until the tracer-controlled slide completes its last pass. After this, limit switch *15* again transmits a command pulse for indexing the template drum. However, drum *11* is indexed (30°) only up to the point where cam dog *6*, set up to the number of passes, actuates lever *7*. At this, lever *17*, mounted on a common shaft with lever *7*, turns through a greater angle than when it is forced out of the slot of the index plate. Because of this, cam dog *13* can operate limit switch *14* which transmits a command pulse for reversing the motion of the piston of the indexing and locking cylinder *1*. For template drum *11* to return to its initial position, a projection of lever *18* must drop behind a projection on lever *7*, holding the latter in the retracted position. As a result, lever *17* cannot drop into any slot of index plate *10*. When, during reverse rotation of the drum, cam dog *4* actuates lever *5*, locking lever *17* is released so that it can drop into the slot of the index plate that corresponds to the initial position.

Limit switch *19* (Fig. 69), operated by cam dog *3*, switches off the tracing slide after the last pass.

4-7. Semiautomatic Six-Spindle Continuous-Type Vertical Chucking Machine, Model 1272 (Krasny Proletary Plant in Moscow)

The model 1272 semiautomatic is intended for turning surfaces of revolution of various configurations under conditions of large-lot and mass production. The high design rigidity of this machine and the amply powered drive enable carbide-tipped tools to be used to their full capacity.

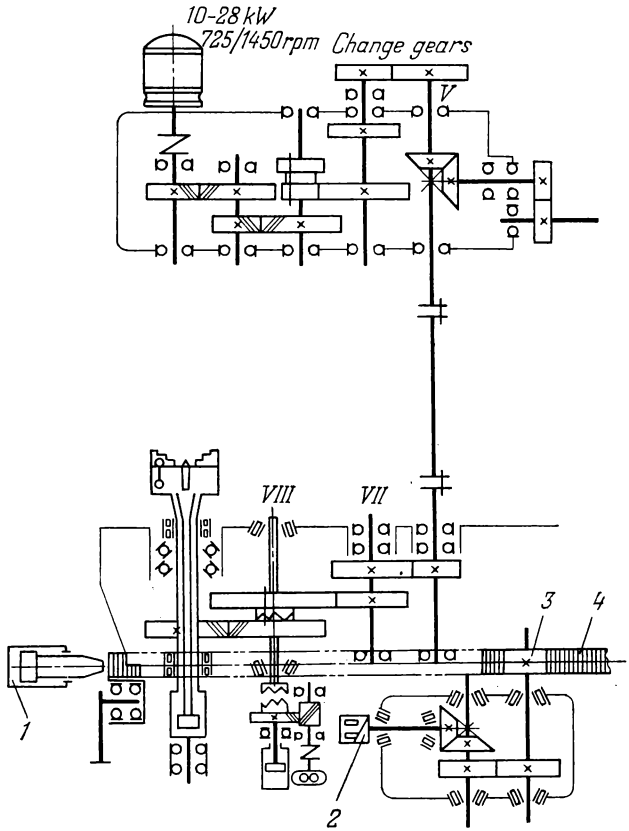


Fig. 71. Kinematic diagram of the spindle drive in the model 1272 semiautomatic six-spindle vertical chucking machine

The application of hydraulic tracer-controlled longitudinal slides increased the production capacity, substituting single-tool machining with automatic readjustment (to compensate for wear) for multiple-tool machining. The semiautomatic may be equipped with special work loading and unloading devices, as well as automatic gauging facilities.

Spindle rotation is powered at each station by an individual drive (Fig. 71) through a reducing gear, change gears and a gear train arranged in the rotary table (spindle carrier). The main drive may incorporate a multiple-speed electric motor and a small-size variable-speed unit providing a range of stepless spindle speeds.

A hydraulic drive is used to power the rotation of the spindle carrier and outer column; it ensures smooth rotation and braking of large masses. Motion is transmitted from rotary hydraulic motor *2* to the carrier ring gear *4* through spur gear *3*. The carrier is locked by hydraulic cylinder *1* having a locking member that enters a tooth space of ring gear *4*. The operation of cylinder *1* is interlocked with that of the carrier rotation mechanism.

Vertical and Horizontal Tool Heads

The vertical head (Fig. 72) is linked to the rod of the vertical-feed cylinder *4* into which oil is admitted through the grooves of two-edge valve *5*.

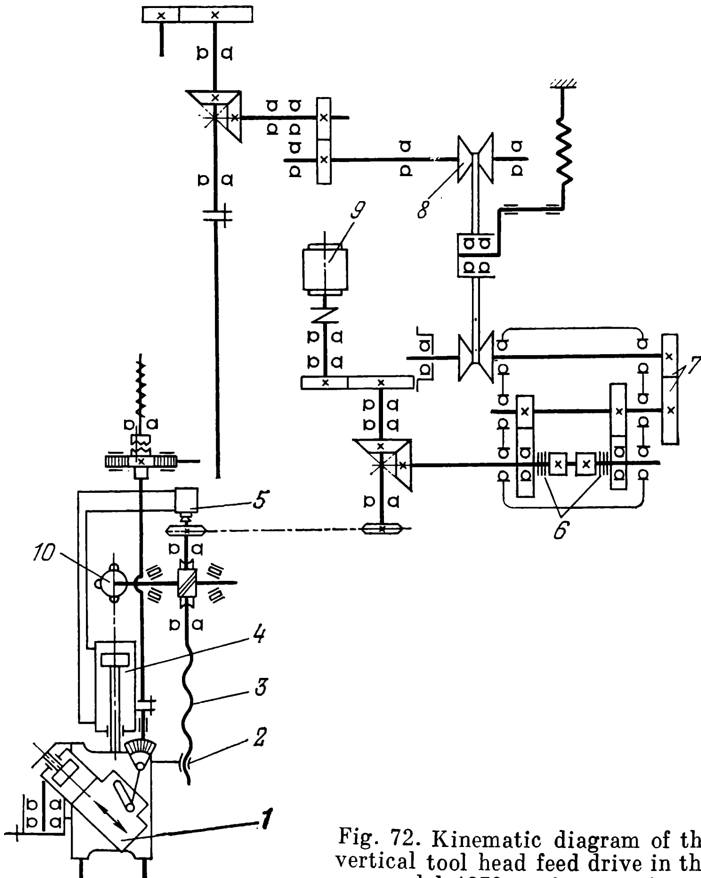


Fig. 72. Kinematic diagram of the vertical tool head feed drive in the model 1272 semiautomatic

The valve is controlled by floating screw *3*. Nut *2* is rigidly linked to the vertical head. Powered by the spindle drive through variable-speed unit *8*, change gears *7*, and two-speed gearbox *6* with electromagnetic clutches, screw *3* is displaced in reference to nut *2* and actuates valve *5*. Being shifted, the valve spool admits oil into the head end of vertical-feed cylinder *4*. This effects working feed of the vertical head and the nut and screw travel away from the valve. However, in its rotation, the screw again shifts the valve spool, etc. Rapid approach and withdrawal of the head are effected by a separate electric motor *9* which is controlled by master switch *10*. At the same time, the master switch engages and disengages the electromagnetic clutches, doubling the rate of feed during the cycle or producing an intermittent feed for chip breaking. Tracer-controlled tool slide *1* is mounted on the ways of the vertical head (at an angle of 60° to the axis of turning) to enable shoulders to be faced on stepped shafts. It reproduces the shape of a flat template with the aid of a two-edge tracing valve (not shown in the diagram). This principle of vertical head operation enables a definite rate of feed per spindle revolution to be obtained. In addition, as in any hydraulic servosystem, leakage in the hydraulic circuit and lack of rigidity in the kinematic characteristic of hydraulic drives are compensated to a certain degree by the tracing (stylus) valve, due to feedback (by means of a lead screw in model 1272), and affect the operation of the head less than they would in an ordinary hydraulic drive.

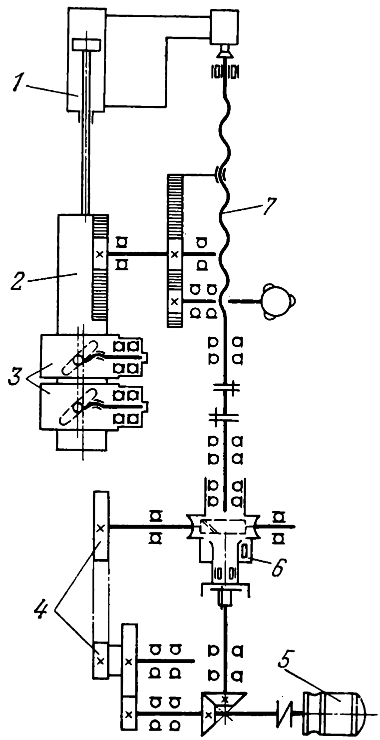


Fig. 73. Kinematic diagram of the horizontal tool head feed drive in the model 1272 semiautomatic

The horizontal head (Fig. 73) operates in a similar manner, except that control screw *7* for working feed and rapid traverse of the head is driven by a separate electric motor *5*, and overrunning clutch *6* is used to join the working feed and rapid traverse gear trains.

Smoother horizontal feed is obtained by an arrangement in which feed cam *2*, linked to the rod of horizontal-feed cylinder *1*, traverses the horizontal tool slides *3*. The required rate of feed is set up with change gears *4*.

Pneumatic and Pneumohydraulic Drives

Because of their vital drawback, considerable nonuniformity of travel due to elasticity of the air and large leakage, pneumatic systems have not found application in the feed drives of automatic machine tools. Pneumatic devices have been extensively employed, however, for powering auxiliary motions which do not usually require uniform velocity, because they substantially reduce the time losses on such operations due to the much higher rates of flow of air (up to 300 m per sec) as compared to oil.

In recent years, pneumohydraulic devices have been widely used to take advantage of the high velocity of idle and handling motions provided by pneumatic systems and the uniformity and smoothness of hydraulic drives. These combined systems utilize the favourable property of viscous liquids to flow at constant pressure (without expansion) from one cavity to another. As a result, the above-mentioned shortcomings of pneumatic systems have been eliminated for all practical purposes.

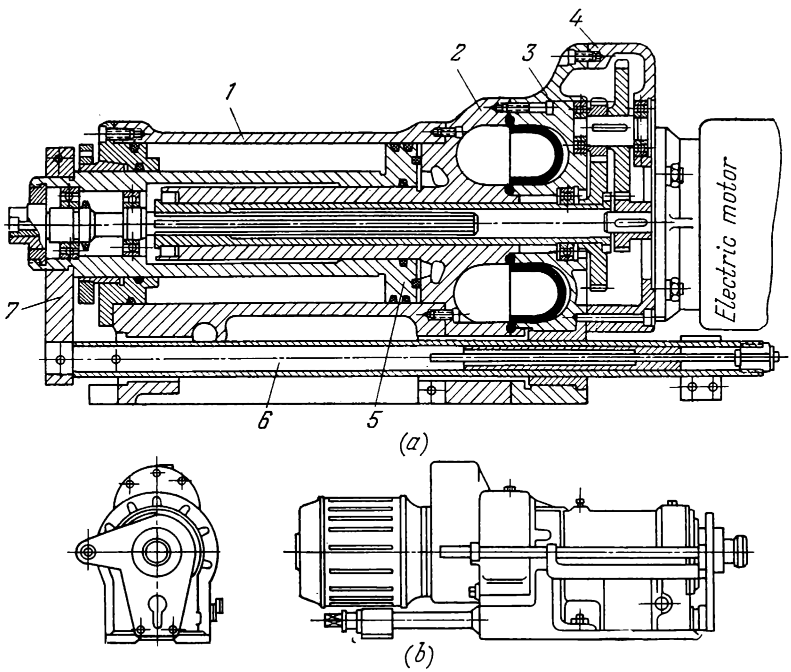


Fig. 74. Axial section of the model ГС-2 pneumohydraulic power unit

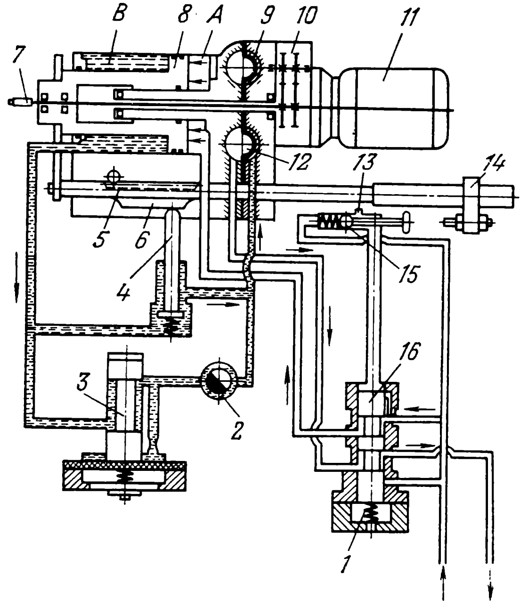


Fig. 75. Pneumohydraulic circuit diagram of the model ГС-2 power unit

As in the case of hydraulic drives, pneumohydraulic systems incorporate in-travel control of the working member.

Figure 74 illustrates the model ГС-2 quill-type pneumohydraulic power unit. It is used in unit-built machine tools and transfer machines for various drilling, boring and milling operations.

In this unit, combined quill and piston *5*, linked by strap *7* to bar *6*, travels along power cylinder *1*. The bar serves as a second guide of the quill and also for mounting various attachments. Housing *2* is secured to cylinder *1* while another housing, *3*, is secured to housing *2*. Housings *2* and *3* have annular cavities between which a special diaphragm of oil-resistant rubber is arranged to separate the compressed air from the oil.

Housing *2* contains a check valve of the hydraulic circuit and a pneumatic directional valve, while in housing *3* there are a flow-control valve, reducing valve and a pulse-controlled valve for automatically reversing quill travel. Reducing gear *4* with change gears is secured to housing *3*.

The pneumohydraulic circuit of the power unit (Fig. 75) provides the following operating cycle: rapid approach; working feed, and rapid return.

Rapid approach. Under the action of spring *1*, the spool of valve *16* is in its upper position. Compressed air (at a pressure of 4 to 6 kgf per sq cm) from

the mains is admitted through valve *16* to end *A* of the power cylinder. Oil from end *B* flows through valve *4*, opened during rapid approach by adjustable cam dog *6*, and into cavity *9*. Air is forced out of cavity *12* through valve *16*.

Working feed. Working feed begins when cam dog *6* runs off the rod of valve *4*. The spring closes the valve and oil flow ceases. After this oil passes to cavity *9* through reducing valve *3* and working-feed flow-control valve *2*. The combined quill and piston *8* travels at the rate of working feed from right to left. Bar *5* travels with the quill.

Rapid return. At the end of quill feed, adjustable stop *14*, mounted on the bar, opens ball valve *15* admitting air to the upper end of valve *16*. This shifts the spool downward to admit air into cavity *12*. At this moment, channel *13* is closed by a solenoid-controlled valve (not shown). Oil forced out of cavity *9* opens check valve *4* and passes into end *B* of the cylinder. Air from end *A* passes out through valve *16*. Combined quill and piston *8* rapidly returns to the initial position.

In repeating the cycle, the solenoid-controlled valve allows the air to exhaust through channel *13* so that the spring shifts the spool of valve *16* to the upper position again.

The drive to spindle *7* from electric motor *11* is set up by means of change gears *10*.

CHAPTER 5

WORKING MEMBERS OF AUTOMATIC MACHINE TOOLS

5-1. Stock and Blank Feeding Mechanisms

Blanks and stock of the following types are handled in automatic machine tools: cold-drawn wire up to 10 mm in diameter; cold-drawn bar stock up to 110 mm in diameter (the use of larger bar stock leads to an excessively high loss of metal as chips); cold-drawn tubing up to 150 mm in diameter; closed-die forgings with the dimensions of the locating surfaces maintained within the 5th grade of accuracy of the USSR Standard (smith forgings require preliminary machining of the locating surfaces); and castings with locating surface tolerances according to the 5th grade of accuracy (castings should not vary considerably in hardness; die castings are convenient blanks).

Mechanisms for Feeding Stock from a Coil

In automatic cutting-off machines with stock fed from a coil, the wire (stock) is held stationary during operation by three chucking devices: rear chuck *2* (Fig. 76), spring collet *5* in a stationary tube at the spindle nose and front chuck *6* which grips the finished but not yet cut-off workpiece at the end of the stock.

The turning operation is performed by single-point tools *4* clamped in cutter head *3*. Cross feed of the tools is effected by cams *7* through levers and longitudinal sliding keys in the spindle.

During the time that the workpiece is being turned, slide *1* with the roll-type wire straightening mechanism travels to the left, away from the spindle, straightening the next length of wire. When turning is completed and the work is cut off, the tools in the cutter head are retracted, the chucking devices open, and straightener slide *1*, travelling to the right, feeds out the wire stock, pushing the finished workpiece out of front chuck *6*. The stock is clamped for feeding out by either a lever- or ball-type gripping device which must overcome large inertia forces of the mass of the coil of wire stock.

A lever-type grip (Fig. 77) is simple in design and dependable in operation. It has been widely employed. Its chief shortcoming is the comparatively deep notch produced by the tooth of the feeding pin.

A ball-type grip (Fig. 78) leaves shallow depressions on the surface of the stock. They are made by the balls which are wedged between the stock and

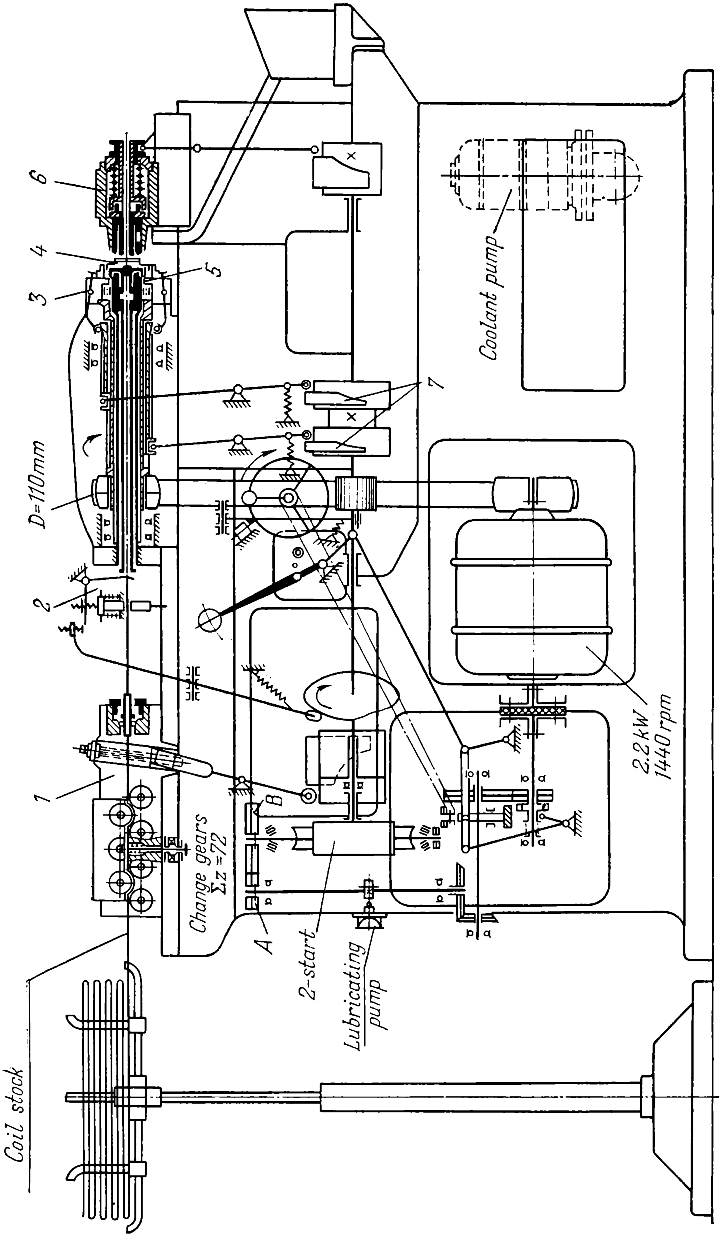


Fig. 76. Kinematic diagram of the model 1106 automatic cutting-off machine

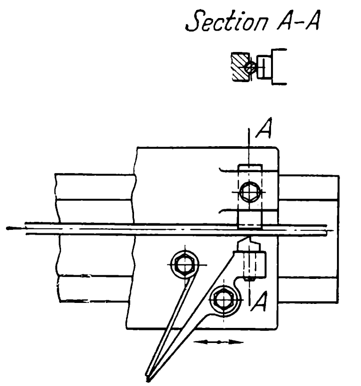


Fig. 77. Lever-type grip for feeding stock from a coil

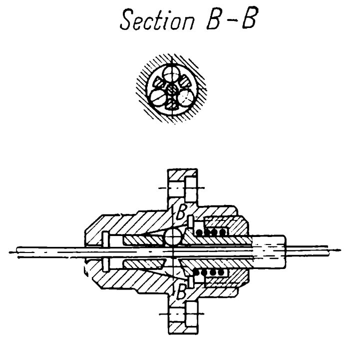


Fig. 78. Ball-type grip for feeding stock from a coil

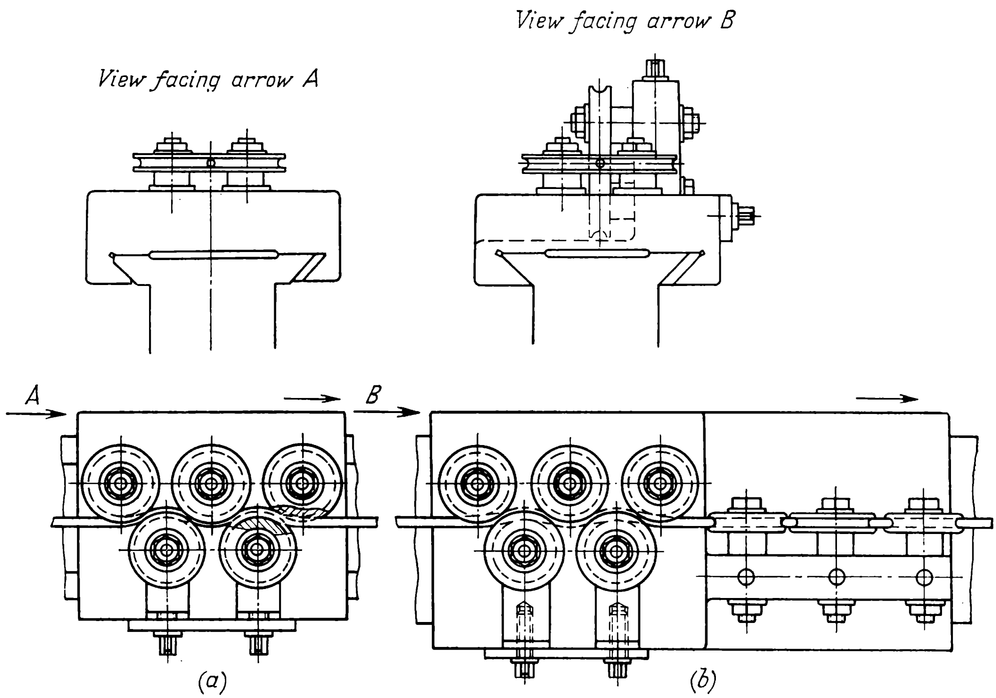


Fig. 79. Mechanism for straightening stock in feeding from a coil: (*a*) single-plane roller type; (*b*) double-plane roller type

the internal taper in the grip body. The depressions are considerably smaller than the notches left by lever-type grips.

Stock straightening, a necessary operation when feeding from a coil, is accomplished by pulling the stock between staggered rollers mounted on a movable slide (Fig. 79).

Double-plane straighteners, with the rolls arranged in perpendicular planes, produce more accurate stock and are more widely used.

Bar Stock Feeding Mechanisms

Bar stock may be fed by one of the following methods:

(1) By gravity. Devices of this kind are simple in design, favourably influence spindle design from the point of view of overall size, but are inconvenient in renewing long bars. Such devices are seldom used and then mainly in multiple-spindle vertical automatic bar machines for small-diameter and comparatively short bar stock.

(2) By driven rolls. This method is used for machining long workpieces.

(3) By advancing the bar stock together with the headstock as in Swiss-type automatic screw machines (see Fig. 27).

(4) By the use of a feeding finger (stock pusher) and tube. This is the main method applied in general-purpose single-spindle automatic screw machines and multiple-spindle automatic bar machines.

Requirements made to pusher-type stock feeding devices include: rapid bar feed at the required moment in the cycle; adjustable length of stock feed; minimum length of the butt end regardless of the adjustment of the length of stock feed; smooth operation without heavy impacts against the stock stop; and provision for automatically stopping the machine when a bar of stock is used up.

In the process of stock feeding, the stock pusher and the pusher tube, in which it is secured, are retracted to take a new grip on the bar, clamped during this period in the chuck. During the return (forward) stroke of the pusher tube, the chuck is open and the stock pusher, gripping the bar stock by spring action, carries it forward by friction. At the end of the feeding stroke, the bar runs against the stock stop, bounces back and is then carried forward again by the pusher which slides along the bar when it bounces back and when the end of the bar is tight against the stop.

To make sure that the stock is fed tightly against the stop, the stroke of the pusher tube and pusher is set up to a larger value, in adjusting the feeding mechanism, than the length of the workpiece to be machined.

Proper operation of the feeding mechanism can be ensured if the pusher does not slip along the bar during the feeding motion except when the bar is being held tight against the stop and during the backward motion to grip

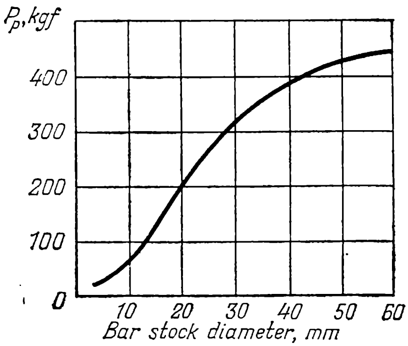


Fig. 80. Relationship between the clamping force of the stock pusher and the stock diameter

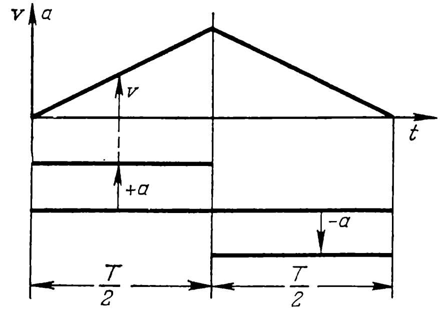


Fig. 81. Law of bar stock motion in feeding with a stock pusher

a new length of bar. This requirement is complied with if the clamping force P_p of the pusher develops a friction force that exceeds the inertia force of the bar. Thus

$$P_p \geqslant k_a \frac{Ga}{gf} \text{ kgf} \tag{23}$$

where G = weight of the bar, kgf
g = acceleration due to gravity, m per sec^2
a = maximum acceleration of the bar in the feeding motion, m per sec^2
f = coefficient of friction
k_a = assurance factor, equal to 3 or 4.

The empirical relationship between the clamping force of the stock pusher and the bar diameter is shown in Fig. 80.

Wear of the stock pusher should be as low as possible.

The required clamping force P_p of the stock pusher can be reduced by designing the mechanism so that the bar stock travels with minimum acceleration a, i.e., with uniformly accelerated motion in the first half of the stroke and uniformly decelerated motion in the second half (Fig. 81). The cam traversing the pusher tube is suitably profiled: a cylinder cam is profiled along a parabola and a plate cam along a logarithmic spiral.

If S denotes the length of stock feed in metres and T the time required for stock feeding in seconds, then (Fig. 81)

$$S = 2 \int_0^{\frac{T}{2}} v \, dt = 2a \int_0^{\frac{T}{2}} t \, dt = 2a \frac{T^2}{2 \times 4}$$

from which

$$a = \frac{4S}{T^2} \text{ m per sec}^2 \tag{24}$$

Stock pushers are made of spring steel 65Г or tool steel У8А and are hardened with the jaws compressed or closed.

Stock pushers with carbide-tipped jaws are also used. They cost 3 or 4 times more but their service life is tenfold that of ordinary pushers.

The length of stock feed is varied either by utilizing only a part of the full stroke or by changing the arm length of one of the levers transmitting motion from the cam to the pusher tube slide (Fig. 82).

For the butt end of the used-up bar to be of constant length, regardless of the adjustment of the pusher tube stroke, the slot along which the slide block of pin *3* (Fig. 82*a*) is moved should be inclined at an angle α in respect to the pusher tube slide. Angle α is equal to one half of the angle of swing of lever *7*.

As pin *3* is adjusted upward by an amount h (Fig. 82*b*), the pusher tube is moved closer to the spindle at the end of its forward stroke by the amount Δ. However, if $\alpha' = \alpha$ the slide block with the pusher tube is moved back, away from the spindle, by the same amount ($\Delta' = \Delta$) as pin *3* is shifted upward with the slide block along the slot by the amount h.

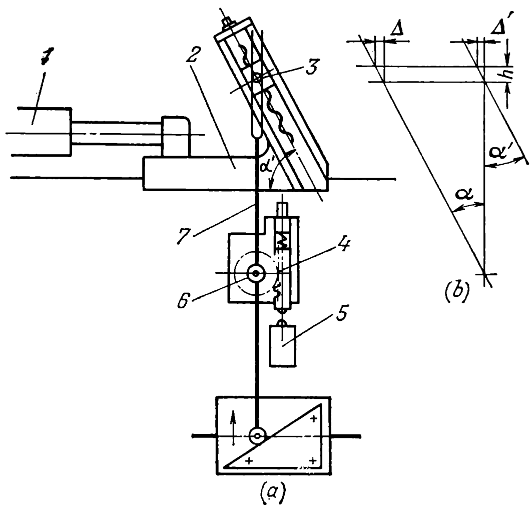


Fig. 82. Diagram of the mechanism used in the model 1A136 automatic screw machine for feeding the bar stock:

1—rear end of the spindle; *2*—pusher tube slide; *3*—pin which is adjusted along the slot of slide *2* in setting up the pusher stroke; *4*—round rack; *5*—limit switch; *6*—pinion linked to lever *7*

When the bar stock is used up and the pusher is moved back to get a new grip on the bar, the tube and slide are not retarded by the pusher, which has slid off the end of the bar, and round rack *4*, under the action of the spring, moves slide *2* farther back than its extreme point in gripping the bar. In its additional downward motion, rack *4* depresses the pin of limit switch *5*. This switches off the electric motor driving the auxiliary shaft which transmits motion to the pusher tube. In this way, the mechanism eliminates the possibility of a breakdown that could occur if the pusher, in the next forward stroke, was to run up against the end face of the butt end of the stock and feed it forward against the stock stop.

Magazine Feeding Devices

The operator fills the magazine, properly orienting the parts, by hand, while they are fed out to the machine tool spindle automatically. The magazine must be filled at periods ranging from 10 to 30 minutes. This enables multiple machine tool handling to be applied if the machining time exceeds 5 to 30 seconds, depending upon the time required to fill the magazine.

Hopper feeding devices are used for shorter machining cycles. Here the parts are dumped into the hopper in bulk without regard to position, automatically oriented by the mechanism and fed out to the magazine chute from which they are automatically fed to the spindle with the required timing.

Magazine feeding is used for the following kinds of blanks: closed-die forgings, smith forgings, castings and semifabricated parts (see p. 115).

The principal parts of a magazine feeding device are the magazine proper (usually a chute) which retains the orientation of the loaded blanks; an escapement which separates the blank being fed out from the successive blanks; and the feeder which feeds out the blank to the spindle.

The blanks are loaded into the chucking device of the spindle and the finished workpieces are ejected by operative mechanisms which are not a part of the magazine feeding device (Fig. 83).

Magazine feeding devices are classified in accordance with the method applied to feed the blanks from the chute (or other retaining member) to the feeder since this has a great influence on the design of the device (Fig. 84).

Most widely used are magazines in which the blanks are fed by gravity along steel chutes with hardened and polished surfaces. The weight of the blank should ensure that it drops into the feeder.

Hopper-type containers are used instead of chutes for blanks that retain their orientation well (blanks with a very large length-to-diameter ratio—shafts and rods, or a very small ratio—disks).

Blanks which cannot be fed by gravity require the application of a force. This may be a weight (rarely used), spring (saves space but is difficult to

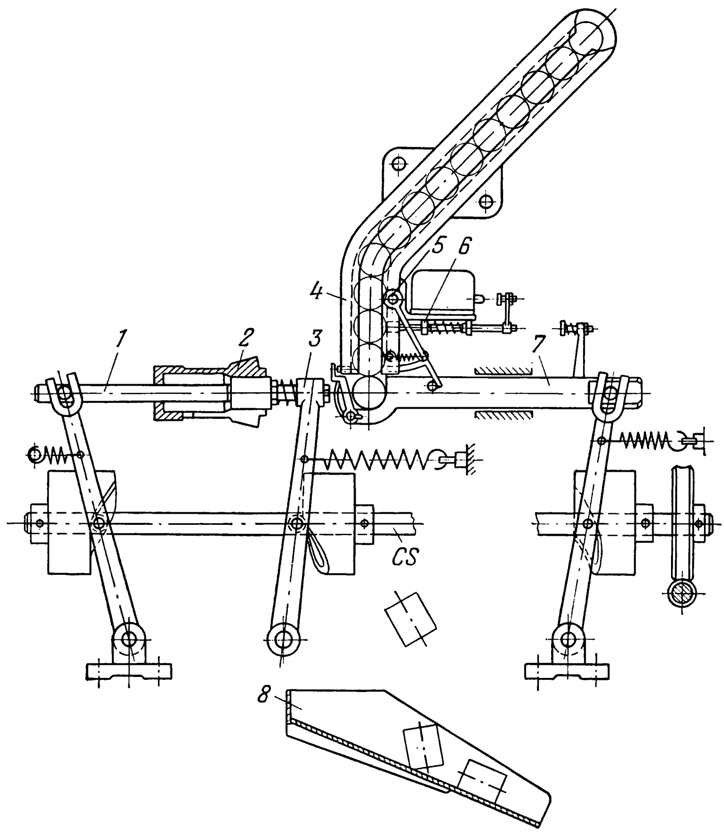


Fig. 83. Magazine feeding device:
1—ejector; *2*—clamping collet; *3*—plunger; *4*—chute of the magazine; *5*—escapement; *6*—device for signalling that magazine is empty; *7*—feeder; *8*—delivery chute

incorporate in the design of the hopper) or friction force (readily combined with continuous feed of the oriented blanks from the hopper).

Chain-type magazines are used for blanks with heads or shoulders, or when additional orientation is required, for example, a plain bushing is to be oriented by a hole in its wall.

In the disk-type magazines, the disks are periodically rotated to present the blanks to the feeder. Magazines with a vertical disk, group VI-C (see Fig. 84), may have very large capacities.

Escapements separate the blank seated in the feeder from the successive blanks. Reciprocating escapements (Fig. 85*a*, *b* and *c*) are used for outputs

of 50 to 70 pcs per min. Rocking escapements (Fig. 85*d*, *e* and *f*) are much simpler in design. Rotary escapements (Fig. 85*g*, *h* and *i*) operate reliably with a high output (250 pcs per min and more) at a low rotational speed.

Feeders feed the blank out of the magazine to the spindle of the machine tool. Frequently, they also serve as an escapement (Fig. 86).

Reciprocating or rocking feeders pass out of the spindle zone after feeding out the blank; this is of substantial importance for single-spindle automatic screw machines. These feeders can operate reliably at outputs not exceeding 60 pcs per min.

Disk and drum feeders (Fig. 86*g*, *h* and *i*) can operate at a high output (up to 300 pcs per min) with a low rotational speed. This low speed ensures that the blanks drop reliably into the pockets of the feeder.

One shortcoming is that the feeder disk is continually in the spindle zone. This restricts the use of such feeders on automatic screw machines. They are widely used, however, in multiple-station automatic machine tools. Their output enables them to be conveniently combined with hopper feeding devices.

A turret hole can be efficiently used as a feeder, the blank being pushed into a holder from a magazine (Fig. 86*l*) arranged behind the turret (in respect to the spindle).

Plungers serve to push the blank from the feeder into the chucking device of the spindle. Plungers are designed with a buffer spring if the chuck has a stop for locating the blank lengthwise (Fig. 87*a*, *b*, *d* and *e*), or they are of rigid construction if the lengthwise position of the blank in the chuck is determined by the final position of the plunger (Fig. 87*f*).

Ejectors may be self-acting, in that they push out the finished workpiece by spring action (Fig. 87*a* and *e*), or they may be of the positive-action type with a drive from a cam (Fig. 87*c*).

Interlocking devices serve to transmit a signal or to stop the machine when the magazine is empty (Fig. 88).

Magazines are usually arranged at the front end of the spindle. The feeder with the escapement is mounted on the rear slide (Fig. 89) or a vertical slide. The motion establishing the timing of magazine operation is transmitted to the feeder by the slide drive mechanism. Thus, the operation of the magazine feeding device is tied in with the general automatic cycle of the machine tool. In automatic screw machines, the plunger is installed in the turret instead of a stock stop.

The mechanism for actuating the pusher tube can be adapted for ejecting the workpiece from the chucking device. This mechanism can also be used for powering the feeder-plunger if the magazine is installed at the rear end of the spindle. In this case, the spindle is full of blanks over its whole length. In the working stroke of the feeder-plunger, the workpiece, machined at one

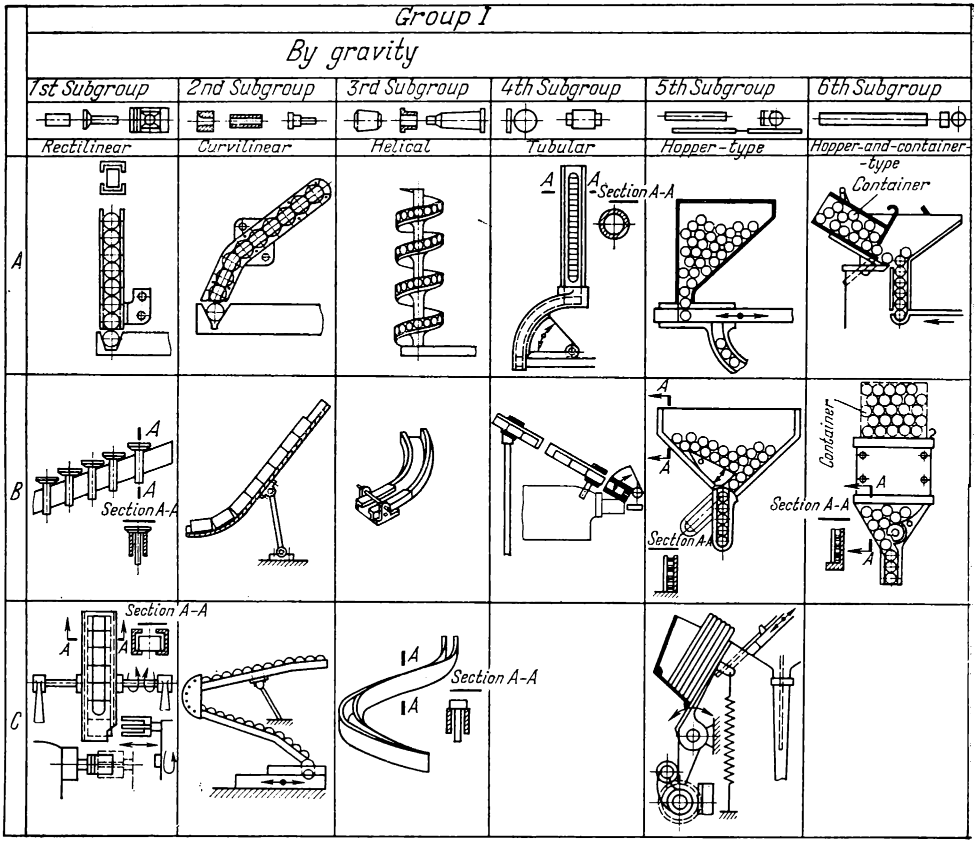


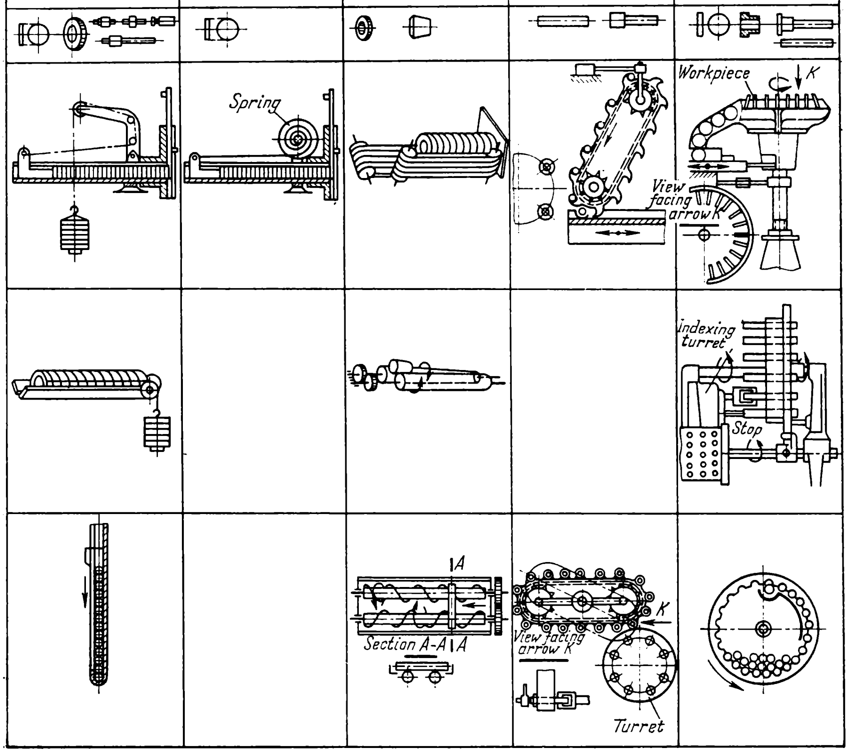
Fig. 84. Magazine feeding device classification according to the method of

end, is pushed out of the chuck and the next blank is fed out to a stop installed in the turret.

In multiple-spindle automatics, the magazine is installed at the front end of the spindle in the loading position. In some cases, the cross slide feed mechanism is used to actuate the feeder while the independent-feed mechanism of the tool spindles is used to operate the plunger.

If supplementary mechanisms are required for a magazine feeding arrangement, they are driven by the camshaft or the auxiliary shaft so that they are kinematically co-ordinated with the general automatic cycle of the machine.

Group II	Group III	Group IV	Group V	Group VI
By a weight	By spring action	Using friction forces	With a chain	With a disk



transferring the blanks from the chute (or other retaining device) to the feeder

Hopper Feeding Devices

If the output of the machine tool is from 50 to 500 pcs per min, blanks of small size, for which the machining cycle is comparatively short, are dumped in bulk into a hopper in which they are automatically oriented and fed out into chute. This chute carries the blanks to the container or magazine. By means of a feeder and plunger, the blanks are taken from the magazine and pushed into the chucking device of the spindle.

Consequently, the purpose of the hopper is to receive the unoriented blanks, to orient them automatically and to feed them out to the delivery chute.

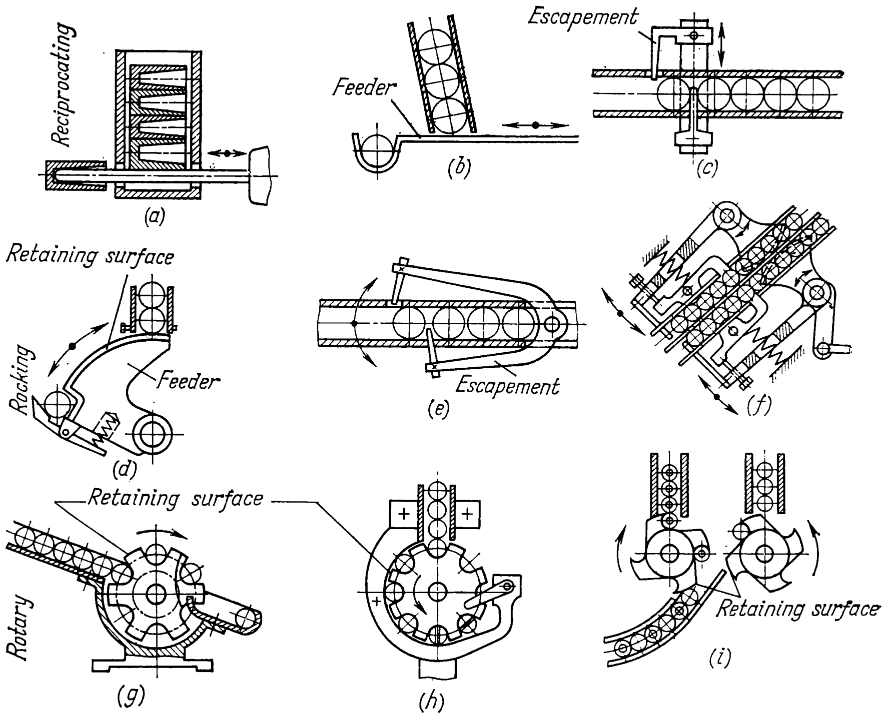


Fig. 85. Types of escapements

To provide the possibility of moving the blank for orientation, it is necessary to separate it in some manner from the remaining mass of blanks. Depending upon the method used to pick out the blanks for orientation, hoppers may be classified as those without catching facilities, and with catching facilities.

Among the hoppers without catching facilities are the vibratory-bowl feeders in which the blanks are picked out and fed along the track, or ramp, of the bowl by inertia forces in conjunction with the friction forces developed by the blank in its motion along the track.

By changing the amperage of the current supplying the electromagnets which vibrate the bowl or by regulating the gap between the core of the electromagnet and the armature, the amplitude of the vibrations can be varied. This, in turn, varies the inertia forces of the blanks and the rate of their orientation. Notwithstanding such control facilities, it is necessary

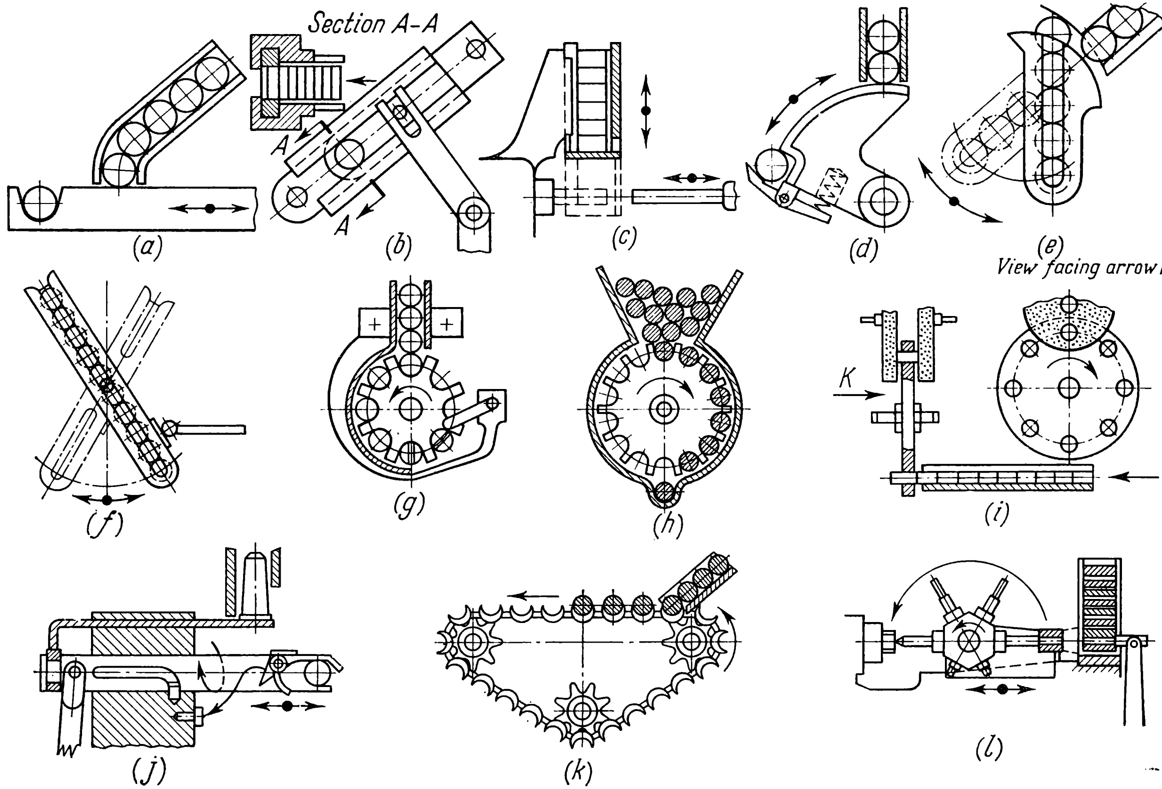


Fig. 86. Types of feeders:
a) (b) and (c) reciprocating feeders; (d) (e) and (f) rocking feeders; (g) (h) and (i) rotary feeders; (j) combined-motion feeder; (k) chain-type feeder; (l) a turret hole serves as a feeder

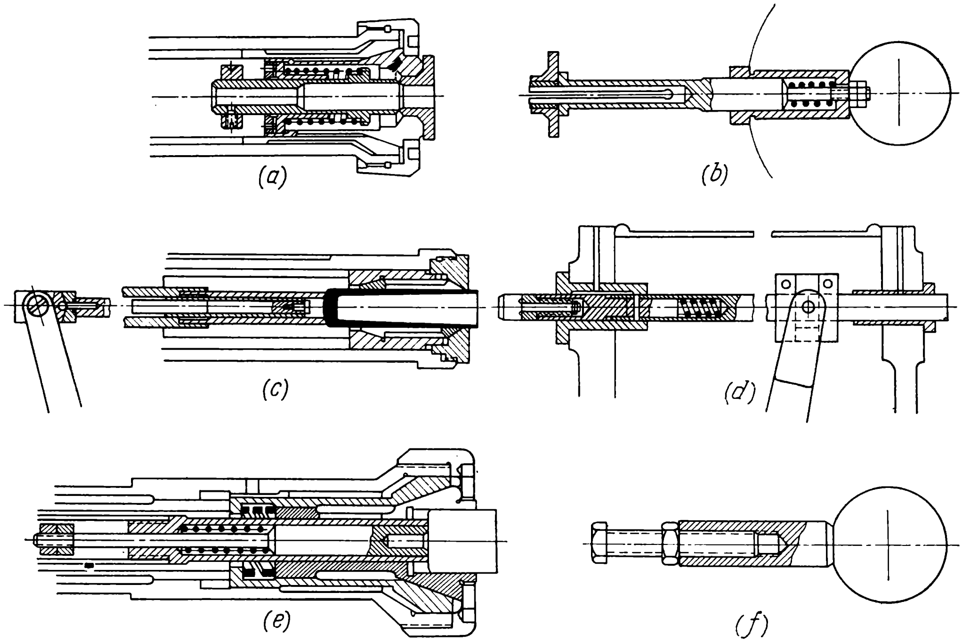


Fig. 87. Chucks, plungers and ejectors

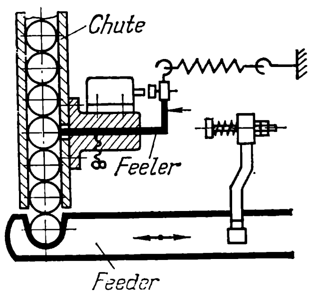


Fig. 88. Diagram of an interlocking mechanism

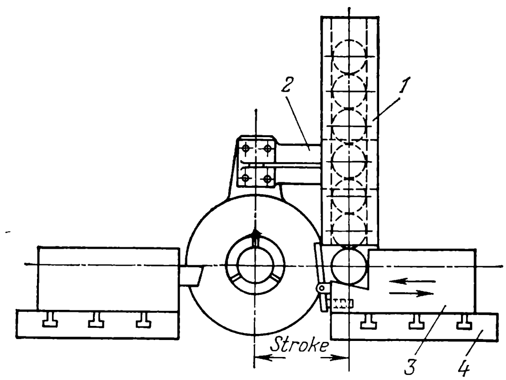


Fig. 89. Magazine mounted on an automatic screw machine:

1—chute; *2*—bracket; *3*—feeder; *4*—rear slide

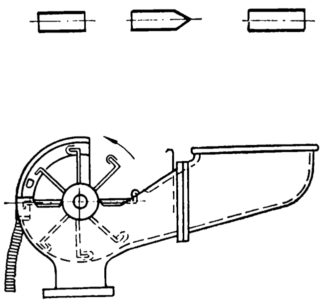


Fig. 90. Hook-type hopper

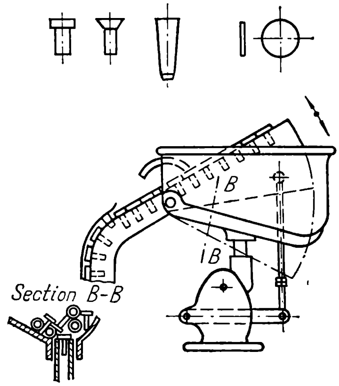


Fig. 91. Centreboard or leaf hopper

to use devices that throw properly oriented blanks back into the bowl if the delivery chute is full.

Vibratory-bowl feeders offer wide opportunities for the orientation of small blanks of the most complex configuration.

In hoppers with catching facilities, the blanks are carried to the orientation zone, to put them into the proper position and to pick out the oriented blanks, by means of (1) the orienting element itself (the pin of the hook-type hopper in Fig. 90) and the slotted leaf in the centreboard or leaf hoppers (Fig. 91) or by (2) displacement of the inclined surface carrying the blanks in the vertical direction and the rolling back of the blanks (rotation about the inclined axis of the disk at the bottom of the hopper, as in Fig. 92), and rotation about the horizontal axis of the drum hopper (see Fig. 95).

In the first case, the orienting element is subject to wear in catching the blank. In the second, blank catching is considerably easier in orientation, and in the drum or tumble orienting device (see Fig. 95) the blank is oriented as it drops into a slot without being caught up first.

Hopper feeding devices are also classified according to the method used to orient the blank. The blank may be oriented by being caught on a hook (Fig. 90), by dropping into a slot (leaf-type) hopper in Fig. 91 and disk-type hopper in Fig. 92, by dropping into a shaped recess (Fig. 93) and by dropping into a tube (Fig. 94).

The output of a hopper depends upon the size of the orienting zone.

In the hook-type hopper, the blanks are oriented at separate points and the output does not exceed 60 pcs per min. In the leaf-type (centreboard) hopper, orienting takes place along a straight line and the output may reach 120 pcs per min. Orientation takes place continuously along the arc of a circle in the disk-type hopper and the output reaches 250 pcs per min at

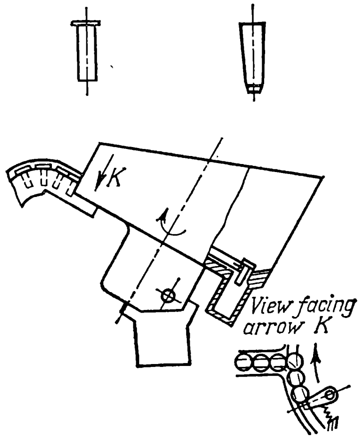


Fig. 92. Slotted-disk hopper

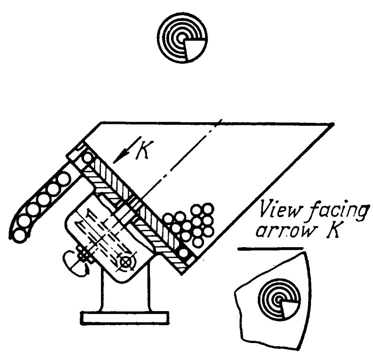


Fig. 93. Disk hopper with shaped recesses in the disk

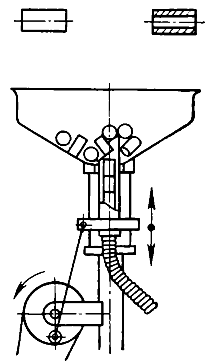


Fig. 94. Hopper with a reciprocating exit tube

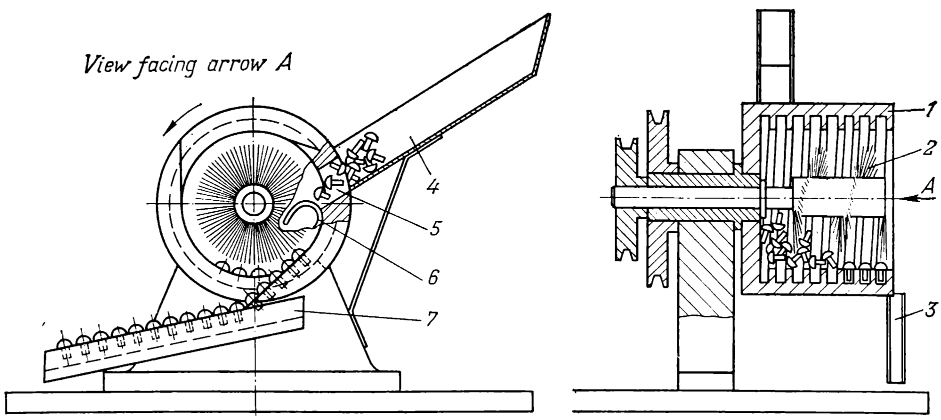


Fig. 95. Hopper with a drum or tumble orienting device:

1—drum with multiple-start internal helical grooves; *2*—steel brush for dislodging unoriented blanks; *3*—delivery chute; *4*—hopper; *5*—input opening in drum wall; *6*—rubber flap for closing input opening when it is at the bottom

a peripheral speed of the disk up to 12 or 14 m per min. In the slot-type drum hopper (Fig. 95) the blanks are oriented on a part of the drum surface and the output may be as high as 800 pcs per min.

Vibratory Loading Devices

Vibratory loading devices have found extensive application for feeding automatic machine tools with separate blanks.

The vibration principle is employed for rectilinear travel of the blanks along a vibrating chute, for elevating the blanks along the helical ramp of a vertical vibratory conveyer with helical oscillation, and for orienting blanks in vibratory-bowl feeders having single- or multiple-start helical tracks, or ramps, on the internal surface of a cylindrical bowl to which helical oscillations are imparted.

The blanks are dumped in bulk into the bowl of such devices (Fig. 96). Under the action of the vibrations, the blanks are dispersed over the convex bottom of the bowl toward the walls where they enter the input end of the track along which they travel upward. The orienting device allows only the properly oriented blanks to reach the delivery end of the track, all others are dislodged and drop back onto the bottom of the bowl.

The oriented blanks form a stockpile at the delivery section of the bowl track and travel along the output chute to the point where the machine tool is to be loaded.

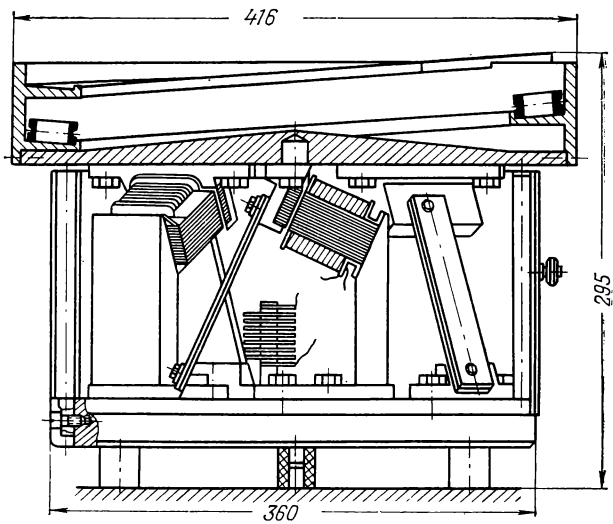


Fig. 96. Vibratory-bowl feeder

The helix angle of the bowl track $\Theta \simeq 2°$ (Fig. 96). The three supports of the bowl, made of spring steel, are inclined at an angle of $\alpha = 25°$ to the vertical. For this reason, three electromagnets, mounted on uprights square to the spring supports, transmit helical oscillations to the bowl with a larger helix angle than that of the track in the bowl. This is required to displace the blanks in the upward direction (see below).

The electromagnets are connected in parallel and are supplied with pulsating direct current (through a common rectifier).

Vibratory loading devices enable blanks to be separated, without the need for catching facilities, for distribution into lines of flow and for efficient orientation even when the features of the blanks, serving as a guide for orientation, are not very prominent or are complicated. They permit blank loading into a machine tool to be automated in cases when no other methods are applicable.

Blank Travel Along the Vibratory Track

A straight chute or track, inclined at an angle of Θ to the horizontal, oscillates at an angle of α to the chute surface (Fig. 97). The chute oscillates according to a sine law. The absolute velocity of the chute in the direction of motion is

$$v_{ch} = A\omega \sin \omega t \text{ m per sec} \tag{25}$$

and the acceleration of the chute is

$$j_{ch} = A\omega^2 \cos \omega t \text{ m per sec}^2 \tag{26}$$

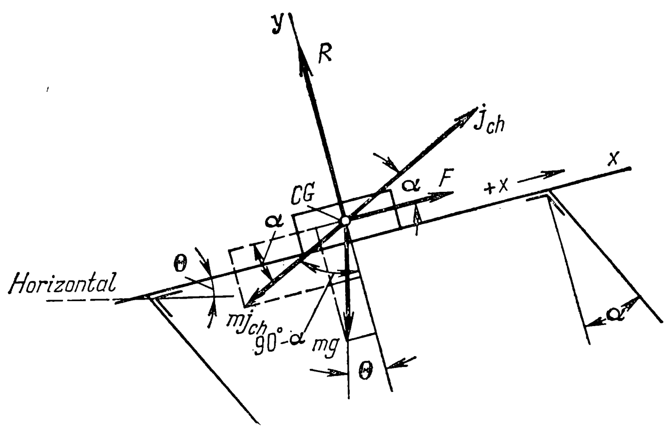


Fig. 97. Forces acting on a blank in a vibratory bowl

while the displacement of the chute from its extreme lower position is

$$S = A(1 - \cos \omega t) \text{ m} \tag{27}$$

where A = amplitude of chute vibration, m
ωt = instantaneous phase angle
ω = circular frequency of the vibration per sec

$$\omega = 2\pi n_m \text{ sec}^{-1}$$

where n_m is the number of oscillations per second.

Acting on the blank (Fig. 97) are the inertia force mj_{ch} (where m is the mass of the blank) in a direction opposite to chute acceleration j_{ch}, weight of the blank mg, bearing reaction R of the chute and friction force $F = Rf$ (where f is the coefficient of friction).

We shall consider the motion of the blank along the chute in a co-ordinate system related to the chute. The sum of all the forces acting on the blank along the x-axis is

$$\begin{aligned} Q_x &= -mg \sin\Theta - mj_{ch} \cos\alpha - Nf(\dot{x}) \quad &\text{at } y = 0 \\ Q_x &= -mg \sin\Theta - mj_{ch} \cos\alpha \quad &\text{at } y > 0 \end{aligned} \tag{28a}$$

The sum of all the forces acting on the blank along the y-axis is

$$\begin{aligned} Q_y &= 0 \quad &\text{at } N > 0 \\ Q_y &= -mg \cos\Theta - mj_{ch} \sin\alpha \quad &\text{at } N = 0 \end{aligned} \tag{28b}$$

The first of the equations in each system, (28a) or (28b), is valid for the case when the blank lies on the surface of the chute; the second—for the case in which the blank leaves the chute surface.

In these equations, N is the normal force exerted by the blank on the chute surface. Thus

$$\begin{aligned} N &= mg \cos\Theta + mj_{ch} \sin\alpha \quad &&\text{at } j_{ch} \sin\alpha > 0, \text{ or at} \\ & &&j_{ch} \sin\alpha < 0 \text{ but when} \\ & &&|j_{ch} \sin\alpha| < g \cos\Theta \\ N &= 0 &&\text{at } j_{ch} \sin\alpha < 0 \\ & &&\text{and } |j_{ch} \sin\alpha| > g\cos\Theta \\ & &&\text{or at } y > 0 \end{aligned} \tag{28c}$$

In the first equation (28a), $f(\dot{x})$ is the coefficient of sliding friction of the blank along the chute surface. This is a function of the sliding

velocity, of the type:

$$
\begin{aligned}
&f(\dot{x}) = f_0 && \text{at } \dot{x} > 0\\
&f(\dot{x}) = -f_0 && \text{at } \dot{x} < 0\\
&f(\dot{x}) = \frac{-mg\sin\Theta - mj_{ch}\cos\alpha}{N} && \text{at } \dot{x} = 0,\ N > 0 \text{ and}\\
& && \left|\frac{mg\sin\Theta + mj_{ch}\cos\alpha}{N}\right| \leqslant f_0
\end{aligned}
\tag{29}
$$

where f_0 is the coefficient of kinetic friction.

Since the forces Q_x and Q_y are counterbalanced by the inertia forces, i.e.,

$$Q_x = m\ddot{x} \quad \text{and} \quad Q_y = m\ddot{y}$$

we can obtain the equation for acceleration of the blank from the equations (28a) and (28b) as follows

$$
\begin{aligned}
&\ddot{x} = -g\sin\Theta - j_{ch}\cos\alpha - \nu f(\dot{x}) && \text{at } y = 0\\
&\ddot{x} = -g\sin\Theta - j_{ch}\cos\alpha && \text{at } y > 0
\end{aligned}
\tag{30a}
$$

$$
\begin{aligned}
&\ddot{y} = 0 && \text{at } \nu > 0\\
&\ddot{y} = -g\cos\Theta - j_{ch}\sin\alpha && \text{at } \nu = 0\\
&\nu = g\cos\Theta + j_{ch}\sin\alpha && \text{at } j_{ch}\sin\alpha > 0 \text{ or}\\
& && \text{at } j_{ch}\sin\alpha < 0, \text{ but when}\\
& && |j_{ch}\sin\alpha| < g\cos\Theta\\
&\nu = 0 && \text{at } j_{ch}\sin\alpha < 0 \text{ and}\\
& && |j_{ch}\sin\alpha| > g\cos\Theta\\
& && \text{or at } y > 0
\end{aligned}
\tag{30c}
$$

Equations (30a), (30b) and (30c), together with equation (29), form a system of differential equations of the travel of a blank along a vibratory chute. They constitute a basis for calculating the parameters of blank motion.

It follows from this formula that the conditions of blank motion vary with the amplitude of chute vibration. At small amplitudes, the blank is moved together with the chute without any relative displacement. An increase in amplitude results in one-sided slipping in the direction of chute elevation, two-sided slipping, and two-sided slipping with bouncing of the blank.

In the helical oscillations of the track in a vibratory bowl, centrifugal forces of inertia act on the blank in addition to the forces shown in Fig. 97. The effect of the centrifugal forces is comparatively small and equation (30) is valid for the helical track of vibratory bowls.

The average speed of motion along a vibratory chute ranges from 50 to 200 mm per sec. The output of vibratory-bowl feeders is lower than the maximum output of mechanical hoppers. It can be increased by using multiple-start helical tracks.

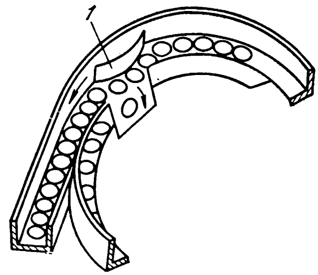


Fig. 98. Pressure break *1* for reducing excess pressure between the blanks

Blank Orientation in Vibratory-Bowl Feeders

The blanks are prepared for orientation by establishing their consecutive single-file travel along the track. This is done by applying one of the following methods: (a) restricting the track width at the very beginning, (b) providing a local narrowed section in a wider (more universal) track, (c) making the track cross section inclined toward the bowl wall so that the blanks are positioned in a line along the bowl wall.

To keep the blanks from climbing up on one another, an inclined baffle (called a wiper) is secured to the bowl wall at a height above the track sufficient to allow one thickness of blanks to pass under and forcing all the upper blanks to drop back into the bowl. Another procedure is to incline the track downward toward the centre of the bowl and provide a small rim that retains only one thickness of blanks. The upper blanks slide off into the bowl.

Excess pressure between the blanks in a row, due to resistances or overfilling of the output chute, may impede orientation and cause the blanks to climb up on one another. This can be eliminated by what is called a pressure break, which is a local projection (Fig. 98) that forces excess blanks off the track when they begin to back up due to the resistance ahead.

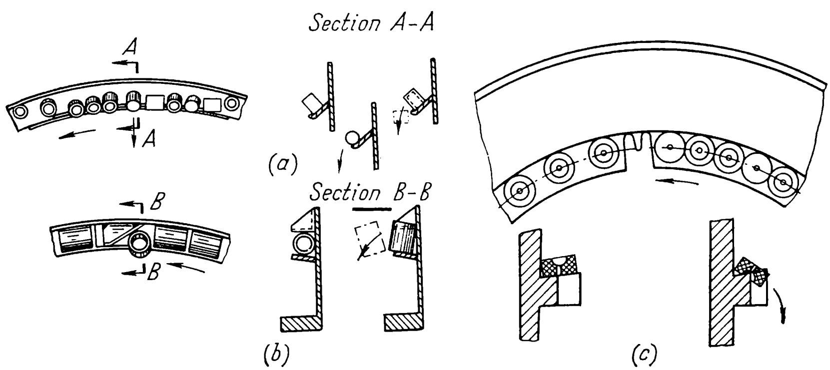


Fig. 99. Blank orientation:
(*a*) for cups; (*b*) for bushings; (*c*) for buttons with a recess on one end

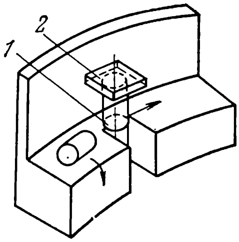


Fig. 100. Orienting cylindrical parts of a length almost equal to the diameter:

1—scalloped dish-out for allowing only vertical blanks to pass and dislodging horizontal ones; *2*—hold-down for retaining orientation of vertical blanks as they pass by the dish-out

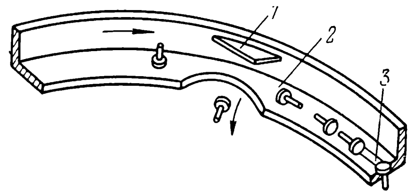


Fig. 101. Orienting device:

1—wiper for dislodging blanks standing on their heads; *2*—dish-out for dislodging blanks located crosswise on the track; *3*—slot for orienting the blanks with the heads upward

To ensure movability of the blanks in orientation, it is sometimes necessary, not only to reduce the pressure between them, but to provide a clearance (for example, in the feed and orientation of small pinions). This can be accomplished by substantially reducing the helix angle of the track just before the point of orientation; in some cases the track may even be inclined downward.

The blanks are oriented by means of orienters which arrange all the blanks in the required position, and selectors which pass only properly oriented blanks, all others being pushed off the track back into the bowl (Figs. 99 and 100).

Since an orienter cannot re-orient blanks from all possible positions, it is usually used in conjunction with another orienter and/or selector. For example, a slot is provided for the body of a headed blank to drop into at the output end of the track (Fig. 101). Before this slot a selector and a local narrowed section are provided which pass only blanks with the body aligned with the track and dislodge all other blanks.

Excess blanks are usually thrown back into the bowl by a pressure break (see Fig. 98) when the output chute is full. If this procedure may damage the blank, a pickup is installed in the gravity-type chute. When the output chute is full, this pickup switches off the bowl vibrator (Fig. 102).

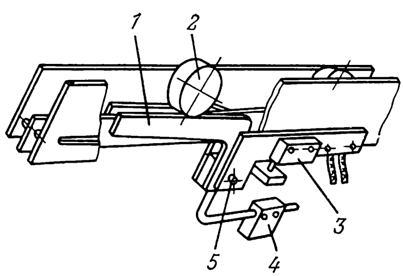


Fig. 102. Pickup which is tripped when the output chute is full:

1—lever; *2*—blank; *3*—small-size limit switch; *4*—counterweight; *5*—pivot of counterweight

Vibrators

Vibratory bowls and chutes are commonly oscillated by electromagnetic vibrators. These are simple and compact in design, reliable and trouble-free in operation because they have no rubbing parts subject to premature wear, or transmitting elements requiring maintenance. Their efficiency is high; they can be quickly started and stopped. They can be readily synchronized in a multiple-vibrator drive.

The ease with which the feeding rates can be varied is a great advantage of electromagnetic vibrators.

In most cases, these vibrators are regulated with a rheostat or autotransformer which is either built into the bowl unit or is arranged on the control desk of the automatic machine tool. Such regulation is to be preferred if the axis of oscillation of the electromagnet armature is located perpendicular to the axis of the spring, and the vibrator has an equal number of electromagnets and springs (see Fig. 96).

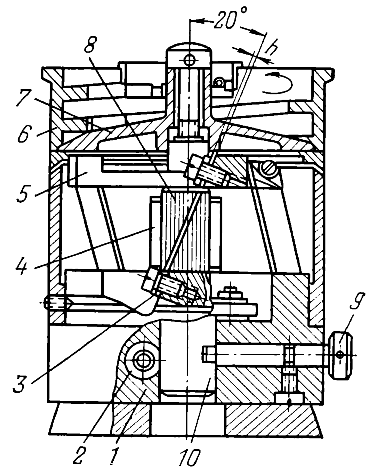


Fig. 103. Vibratory bowl with an adjustable air gap:
1—base; *2*—clamp of the electromagnet upright; *3*—mounting of the spring; *4*—spring; *5*—disk; *6*—bowl; *7*—bottom of bowl; *8*—electromagnet core; *9*—shaft with eccentric pin for air gap regulation; *10*—upright of the electromagnet

In small vibrators (Fig. 103) the rate of blank feed is varied by regulating the air gap between the core of the electromagnet and the armature. This can be done simply, without complicating the design, only in vibrators having a single electromagnet (Fig. 103) and a vertical axis of oscillation. Air gap regulation is less convenient and accurate than regulation by means of a rheostat or autotransformer.

5-2. Chucking Devices

Depending upon the degree of automaticity of the machine tool, automatic, semiautomatic or manual chucking devices are employed.

An *automatic chucking system* is used in automatic bar machines and in chucking machines with magazine feeding facilities. Collet chucks of various types are most widely employed for clamping the stock in lathe-type machine tools.

A collet chuck should satisfy the following requirements: it should provide concentric clamping with a radial runout not exceeding 0.01 to 0.04 mm; reliable holding of stock with normal size errors (within the limits of 5th

grade accuracy, according to the USSR Std); constant length of bar feed; constancy of the elastic properties of the collet and high wear resistance of the jaws (collets should clamp from 50,000 to 100,000 blanks during their service life).

Collets are made in the Soviet Union of tool steel У8А, У10А or spring steel 65Г. The tapered surface is hardened to 58-62 R_C, and the shank to 40-45 R_C.

The jaws of collets are expanded slightly in hardening so that the bar stock can be freely fed out when the collet is open. The included angle of the taper is usually 30°. Collets have from two to six slits depending upon the diameter of the bar or tube stock. Thus:

Stock diameter, mm	<20	20 to 60	40 to 100	60 to 150
Number of slits in collet	2	3	4	6

In machining separate blanks, the diameter of the clamped part of the blank should not be less than one half of the diameter of the largest machined surface.

Distinction is made between three principal types of collets (Fig. 104): push-out type with a front taper; drawback type with a back taper, subject to tensile forces in clamping the bar; and stationary type with a back taper in which the front ends of the jaws bear against a ring screwed on the spindle nose.

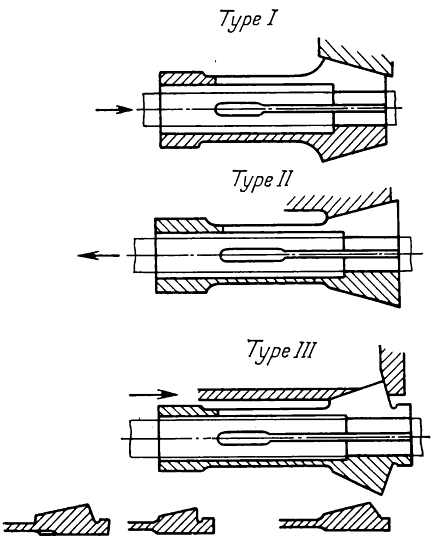


Fig. 104. Principal types of collets

The third type of collet is closed by a sleeve which is pushed forward against the taper of the collet.

The advantages of the *push-out collet* are the simplicity of its construction and the simple joint between the collet tube and the collet.

However, these collets have essential shortcomings such as the possibility of the collet being accidentally closed as it is carried forward by the bar being fed; the tendency of the axial component of the cutting force to open the collet (by pushing it back) allowing the stock to slip back; inaccurate location of the collet by the thrust nut; and a lack of concentricity in clamping that may result from the deformation of the collet jaws due to the longitudinal load.

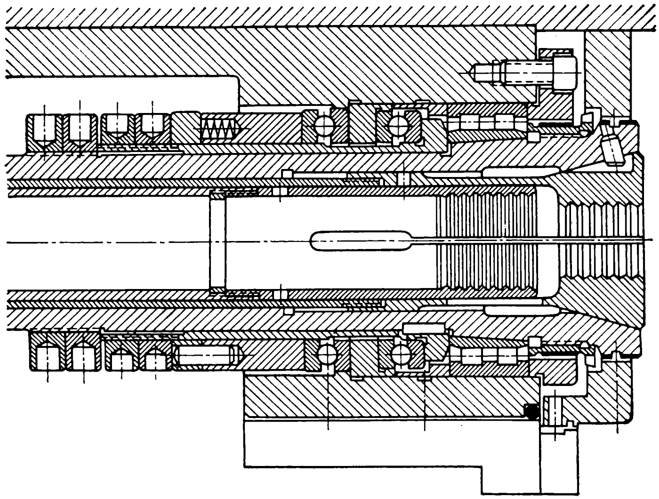


Fig. 105. Spindle nose with a drawback collet

Because of these features, push-out collets with a front taper find no application in modern machine tools.

Chucks with a *drawback collet* having a back taper (Fig. 105) have the following advantages: the collet taper fits into the tapered seat in the spindle, ensuring proper centring of the stock or blank; jamming during stock feed is excluded; the axial component of the cutting force tends to increase the clamping force; the jaw members are stretched and not compressed in clamping so that the collet and its taper are deformed only slightly; this collet chuck has a small radial overall size. The last feature is the reason for the wide application of such collets in multiple-spindle automatics because of the more or less crowded arrangement of the spindles in the carrier.

Shortcomings of drawback collets include: wear of the tapered seat in the spindle; relative weakness of the screw joint between the collet and its tube (ruptures of the tube or collet sometimes occur at this point if the clamping force has not been properly adjusted); the bar is pulled away from the stock stop as it is being clamped.

The elasticity of the members transmitting the pulling force to the collet greatly affects chuck operation.

If the elasticity of the clamping action is not very high (as in Fig. 106, where the elastic elements are the fingers in the holder), the chuck can operate dependably at variations in stock diameter within the limits of 5th grade accuracy.

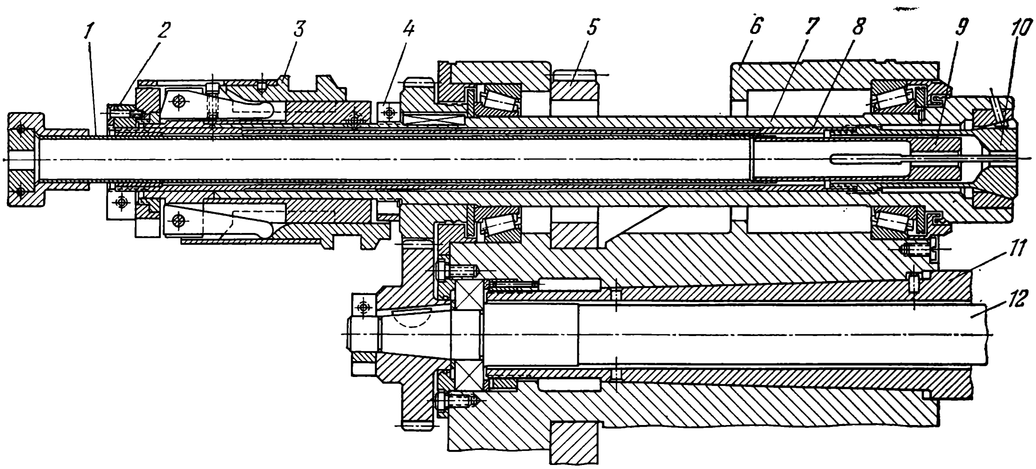


Fig. 106. Chucking device with a drawback collet:

1—pusher tube; *2*—nut for regulating the clamping force; *3*—finger holder; *4*—nut for adjusting the spindle bearings; *5*—gear for indexing the spindle carrier; *6*—spindle carrier; *7*—spindle; *8*—collet tube; *9*—stock pusher; *10*—collet; *11*—carrier stem; *12*—central shaft

If the pulling force required to operate the chuck is transmitted through a spring (Fig. 107), the permissible diameter tolerance on the stock can be increased and, with a proper selection of the spring stiffness, may reach 0.3 and even 0.5 mm.

Chucking devices with a *stationary collet* (Fig. 108) possess the following features: precise bar feed since the collet is not displaced axially in clamping; no threaded joints between the chucking tube and sleeve and between the sleeve and collet that are frequently the weakest point of the construction; in clamping, the force is transmitted directly to the tapered head of the collet without applying a load along the whole length of the jaws or deforming the collet.

The main shortcoming of these chucks is the large radial overall size, not allowing it to be applied in multiple-spindle automatics.

Unlike the construction shown in Fig. 106, where the elastic members in transmitting the clamping force to the collet are the inserted fingers of the holder, the chuck illustrated in Fig. 109 has a clamping force developed by springs *4* which are preloaded between collars *2* and *3* by nuts *1*. These nuts are screwed on the end of the spindle and are used to regulate the clamping force.

When collet *15* is open, collars *2* and *3* are held together by screws. In the rotation of the drum cam, sleeve *7* is shifted to the right and balls *8*, forced between sleeve *6* and ring *9*, compress springs *4*.

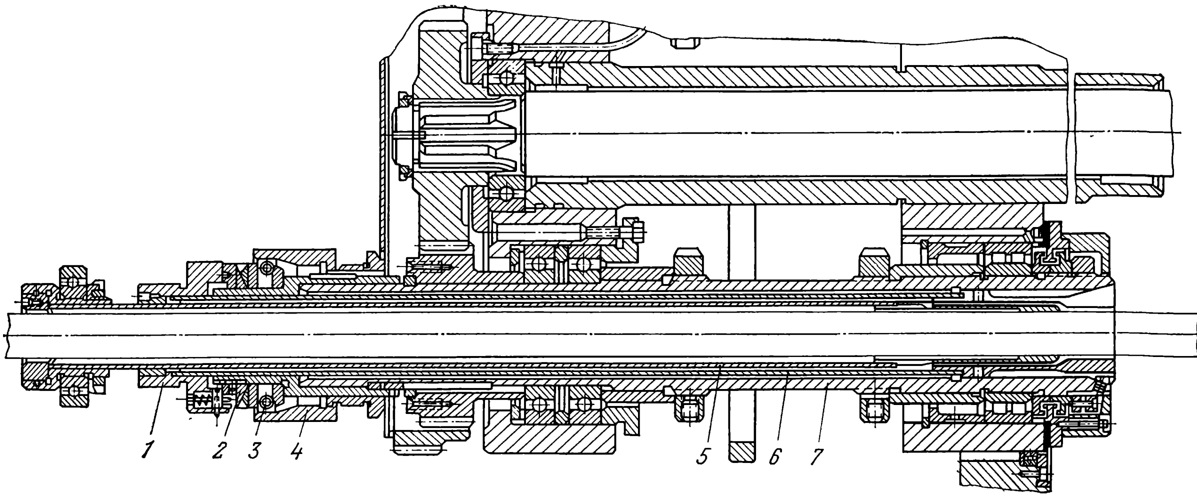


Fig. 107. Chucking device with the clamping force transmitted through a spring:
1—nut for regulating the clamping force; 2—Belleville springs; 3—barrel-shaped roller; 4—sliding sleeve; 5—pusher tube; 6—collet tube; 7—spindle

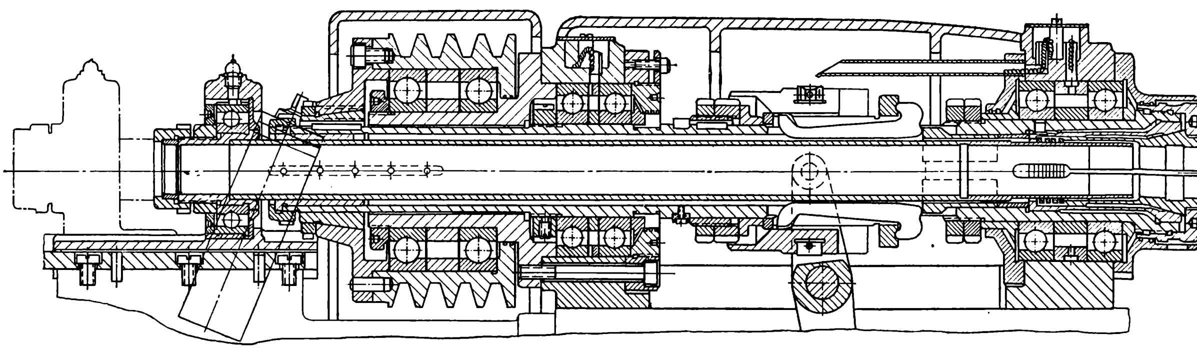


Fig. 108. Chucking device of the model 1A136 automatic screw machine

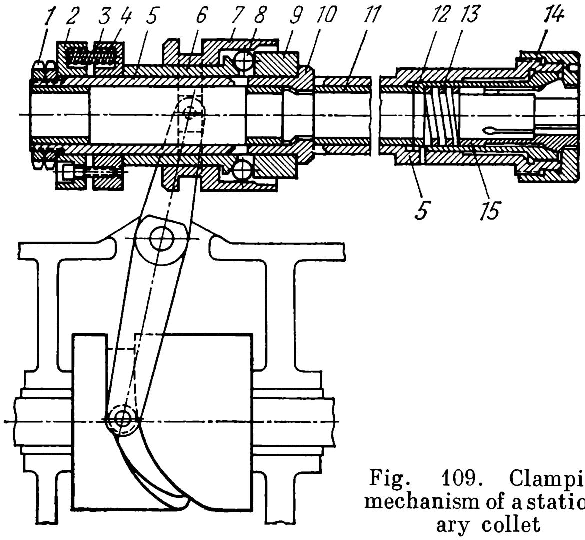


Fig. 109. Clamping mechanism of a stationary collet

The force of the compressed springs is transmitted through balls *8*, ring *9*, sleeve *10* and tube *11* to sleeve *12*. The last sleeve is pushed forward upon the taper of collet *15* and also compresses spring *13*. The longitudinal clamping force, acting on nuts *1* and, through the collet, on nut *14*, is contained within spindle *5*. When sleeve *7* is shifted back, balls *8* move outward, springs *4* are held in the preloaded position by the screws in collars *2* and *3*, and spring *13* pulls clamping sleeve *12* from the collet, thereby releasing the bar.

In a *semiautomatic chucking device*, the blank is loaded into the chuck, the finished workpiece is removed and the chucking device is controlled in clamping and unclamping by the operator.

The layout and design of up-to-date semiautomatic machine tools make provisions for the use of loading devices and automatic control of the chucking device if the machine is built into an automatic transfer machine or line.

A semiautomatic chucking system consists of the clamping mechanism, or chuck, and the operating mechanism which transmits the clamping force to the chuck.

The following types of chucks are employed: collet, jaw and expanding-pin arbors with wedge-type keys.

Various kinds of pulling devices are used. In a semiautomatic six-spindle chucking machine, the pulling force for clamping is produced by a spring with a low force-deflection ratio (spring rate), as in Fig. 110, acting on the drawbar. The tapered head of the drawbar actuates the chuck levers, closing the jaws and clamping the blank. At the loading position, a cam-operated rod

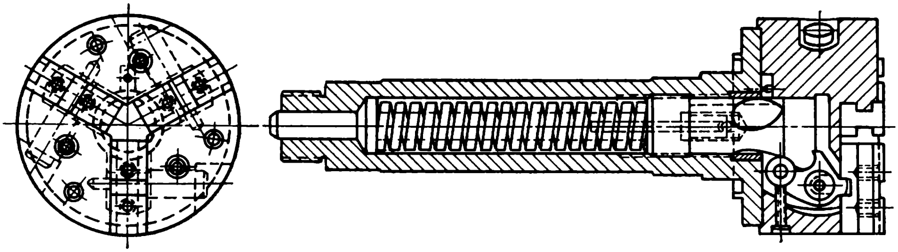


Fig. 110. Spring-operated jaw chuck

depresses the end of the drawbar, compressing the spring and spreading the chuck jaws. This releases the finished workpiece and allows the next blank to be loaded into the chuck.

The pulling force for clamping is regulated by a nut screwed into the spindle bore. Adjustment of this nut may prove difficult in the given version of chuck construction. The difficulty can be eliminated by changing the method of fastening the head to the drawbar so that the nut can be adjusted by turning the drawbar.

The construction of this chucking device and its control are simple. The device is not suitable, however, for heavy-duty jobs in which a large clamping force is required.

Single-spindle semiautomatic lathes and multiple-spindle vertical chucking machines for heavy-duty jobs, with cam-actuated tool slides, use mechanically operated chucking devices because of the greater reliability of such operating mechanisms. The pulling force for clamping is developed by a power screw and nut.

The model 1282 semiautomatic eight-spindle vertical chucking machine, intended for large-lot and mass production, has cam-actuated tool heads. Cyclic operative mechanisms with a mechanical system of operating cycle control are used for the auxiliary working members. The chucking device is also mechanically operated (Fig. 111). The pulling force is transmitted to the chuck from nut *1* of power screw *2* resting on split thrust bearing *3* which is secured in the bore of spindle *4*.

The power screw is driven by an auxiliary vertical shaft through two pairs of bevel gears and a reversing unit consisting of two pairs of spur gears, engaged alternately by double jaw clutch *9*. The jaws of this clutch have inclined sides which automatically disengage the clutch upon overloads. From vertical shaft *8* and hollow shaft *7*, rotation is transmitted to the power screw through gear *6*. At the end of spindle carrier indexing, when one of the spindles enters the loading station, cam *10*, secured to the carrier, pushes

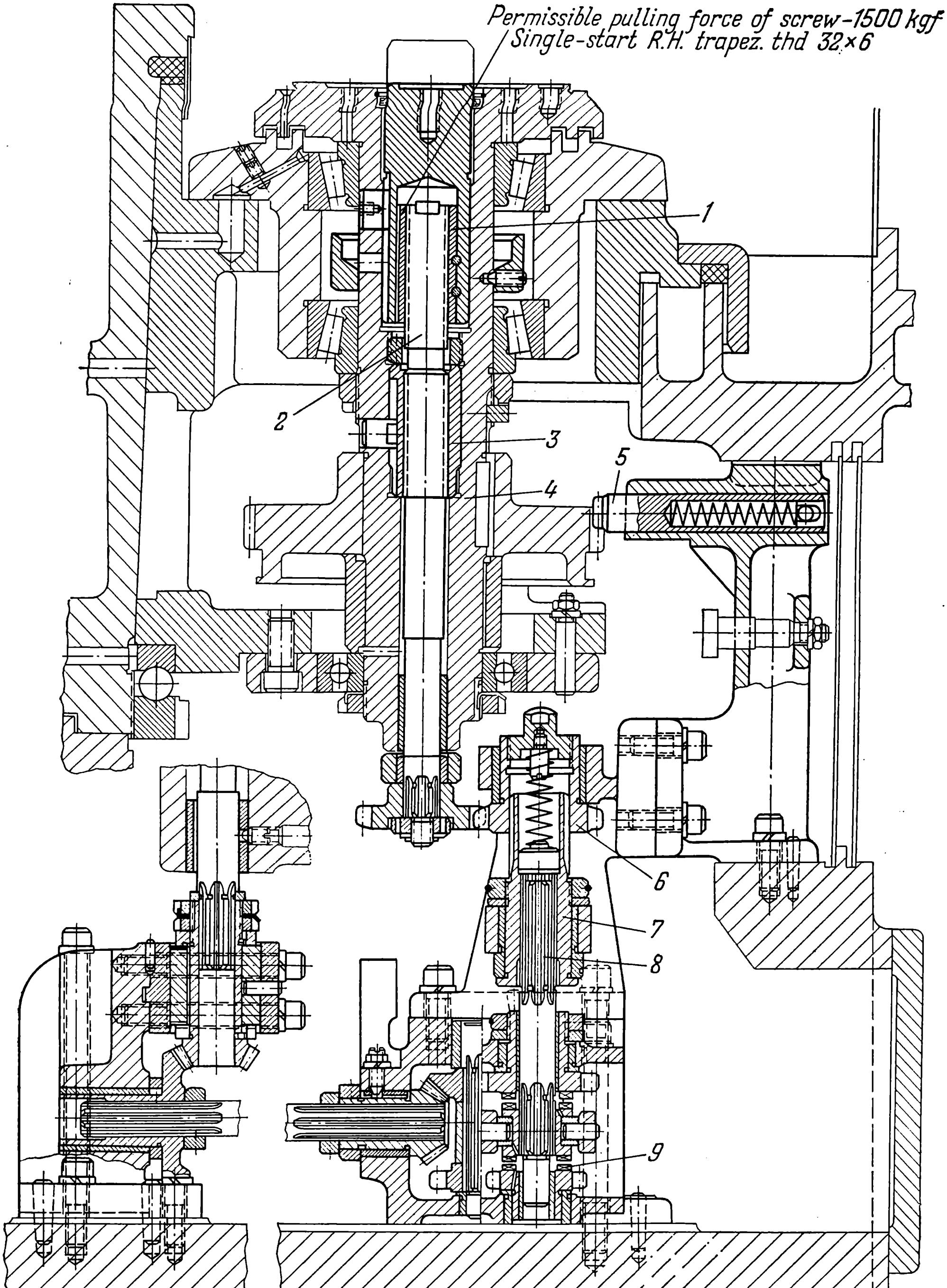
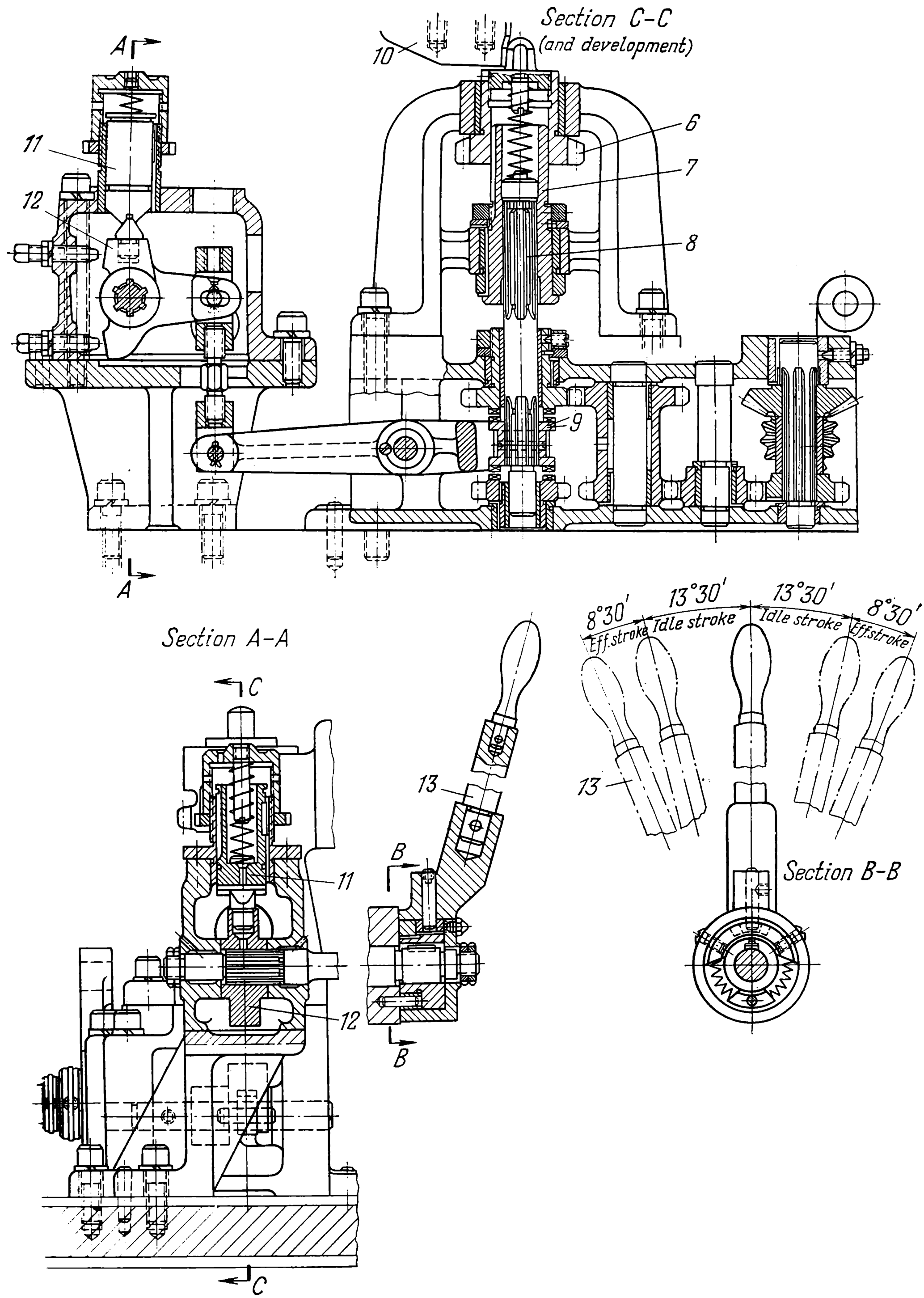


Fig. 111. Semiautomatic power-driven mechanical chucking device



Section C-C
(and development)
A
10
11
12
6
7
8
9
A
Section A-A
C
11
12
C
13
B
B
8°30′
13°30′
13°30′
8°30′
Eff.stroke
Idle stroke
Idle stroke
Eff.stroke
13
Section B-B

gear *6* downward to prevent it from striking the power screw gear. After the carrier is locked, gear *6* slides into mesh, being pushed upward by a spring.

At the loading station the spindle is held stationary by locking member *5* which enters a tooth space of the spindle gear.

When control lever *13* is turned by means of lever *12*, tie-rod and a rocker arm, clutch *9* is engaged for either forward or reverse rotation (to clamp or release the workpiece). In its motion, the tooth of lever *12* is subject to the action of the spring of locking member *11*. When the slot in the locking member engages the apex of the tooth, the clutch is held in the neutral disengaged position. As lever *12* is turned in either direction, its tooth becomes disengaged and at the moment its apex comes out of the slot, the locking member, acting through its bevelled side on a bevel of the tooth, turns lever *12* rapidly in the same direction to engage the clutch. At the end of the clamping (or unclamping) action, the chuck jaws and the power screw run up against a positive stop, the clutch is overloaded and is shifted, by means of its inclined jaws, toward the neutral position, overcoming the force exerted by the spring of locking device *11*.

At this, the apex of the tooth enters the slot of locking member *11* and, due to spring action, lever *12* is turned to the neutral position, as is control lever *13*. The blank remains clamped due to the self-locking feature of the power screw and its nut. The clamping force is varied by adjusting the spring of locking member *11*.

The sides of the slot in the locking member make an angle of 71°. Its outer side surfaces make an angle of 90°.

The shape of the jaws with inclined sides on the middle clutch member and on the engaged gears is shown in Fig. 112 as a development on the outside diameter of the jaws. The active surfaces of the clutch jaws are helical to facilitate self-disengagement of the clutch upon overload.

The locking member is made of wear-resistant steel ШХ15, the tooth of carburizing steel 20X, and the jaw clutch member and the gears with which it engages are made of alloy steel 18ХГТ. All of these parts are heat treated to obtain a hardness of 58-62 R_C on the working surfaces.

Blank clamping is not interlocked with carrier indexing. Consequently, the operator must check whether the blank is properly clamped before engaging carrier indexing. This is usually necessary in practice, even if an interlocking device is available.

The model 1К282 progressive-action eight-spindle semiautomatics have a hydraulically operated chucking device (Fig. 113).

The chucking device (jaw chuck or expanding arbor) is fastened to the flange of spindle *8* and is actuated through drawbar *7* and rod *5* by pistons *6* and *10* of two identical cylinders *9* and *11*, arranged in tandem. The cylinders are secured in a bore of the spindle by flanged bushing *1* and sleeve *12*.

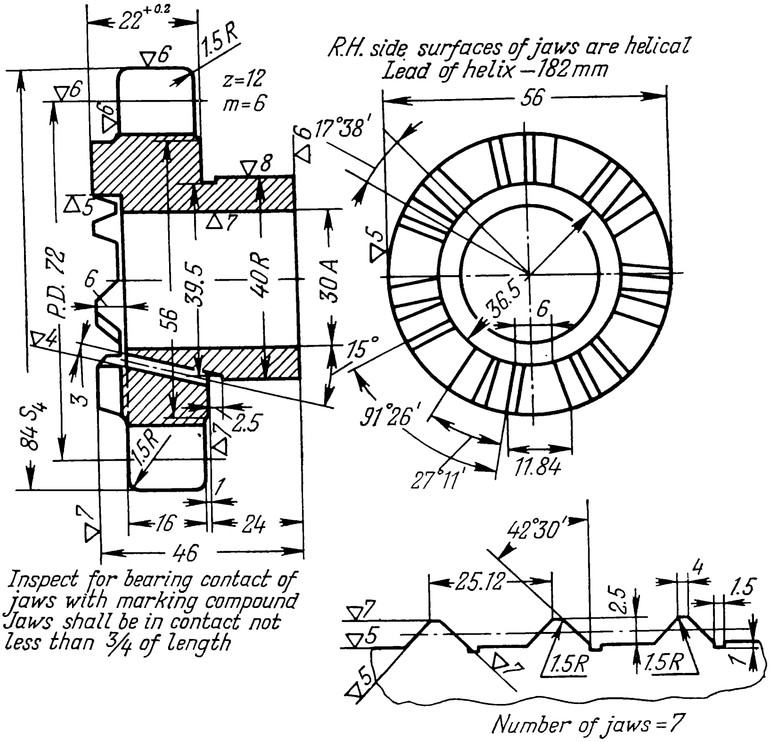


Fig. 112. Clutch member

The chucking device may be self-locking. In this case, the actuating force in unclamping should be greater than in clamping the blank to ensure dependable operation. This is achieved by delivering oil to the head end of the lower hydraulic cylinder in unclamping the blank, and to the rod end in clamping it.

When the spindle is in the loading station, oil enters through port *14* in unclamping, passes upward through the drilled channel in the stationary bar *3*, groove, port and slot of bushing *2*, clearance between flanged bushing *1* and sleeve *12* into the head end of lower hydraulic cylinder *11*. Oil enters the lower end of the upper cylinder through the spline-type channels in rod *5*.

In clamping, oil is admitted through port *13* and passes through ports, grooves and slots in parts *3*, *2* and *1*. From here it passes through clearances between the spindle and sleeve *12*, lower cylinder *11*, partition *4* of the cylinders, and upper cylinder *9* into the upper rod ends of both cylinders.

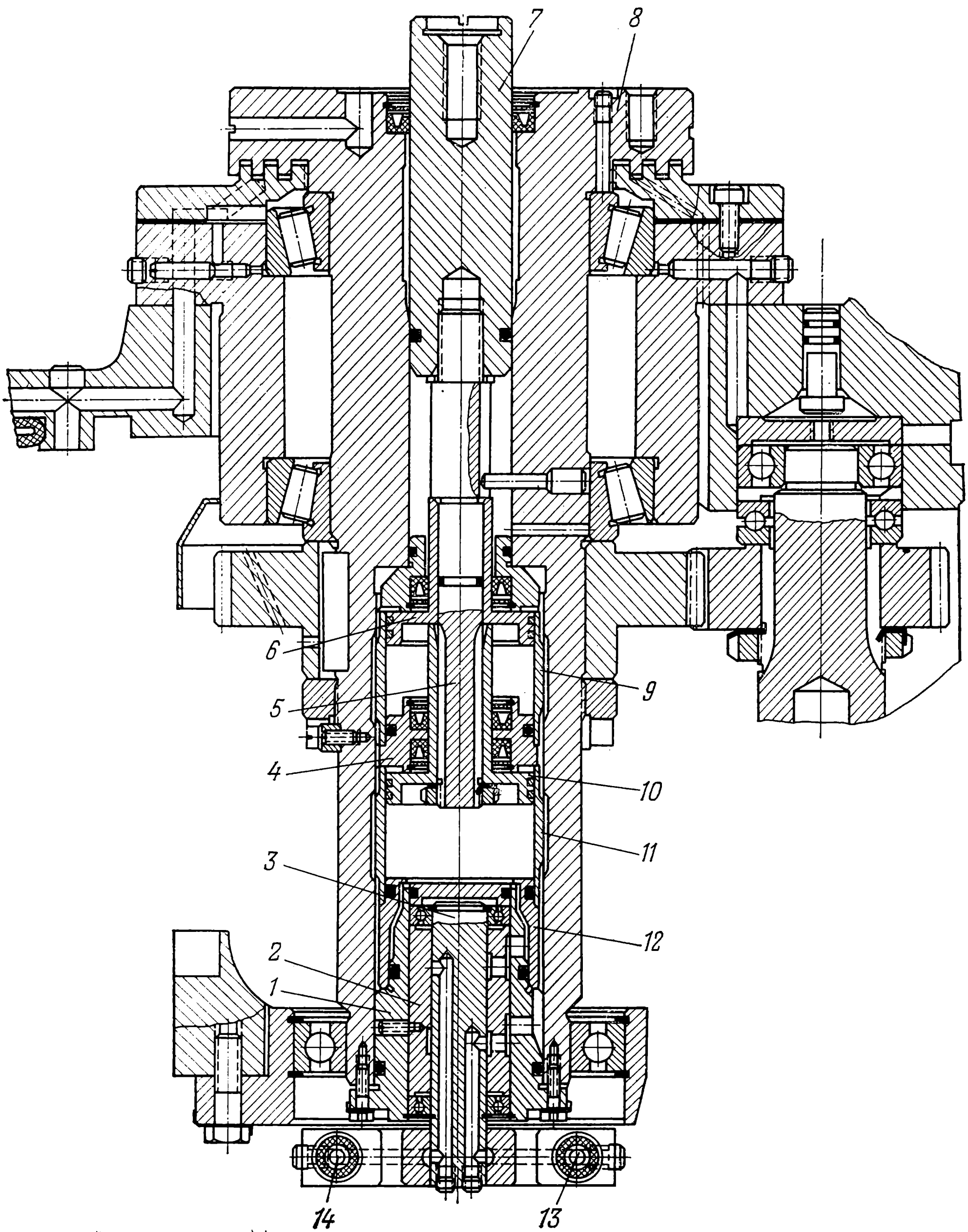


Fig. 113. Hydraulically operated chucking device of the model 1K282 vertical semiautomatic chucking machine

A pressure switch, operated at the end of the blank clamping operation, interlocks the electric control circuit of carrier indexing so that the table can be indexed only after properly clamping the blank.

The oil pressure in the upper end of the cylinders is continuously maintained during indexing and at the working stations of the spindle, keeping the blank firmly clamped.

5-3. Tool Slides of Automatic Machine Tools

Tool slides are the principal working members of automatic machine tools of the lathe type. Hence, the types of tool slides, their number, arrangement, motion cycles and operative mechanisms (actuating devices) of their drives are determined by the volume of production and the manufacturing process for which the automatic machine tool is intended.

On account of the aforesaid, tool slides will be considered separately for each group of automatics.

Tool Slides of Single-Spindle Automatic Screw Machines

These tool slides are classified according to the following features:

1. In respect to the *direction of travel* in reference to the spindle and work axes as end-working (usually turret), cross and compound slides. Compound slides have both longitudinal and cross motions and have been used in certain new models for straight turning behind a shoulder, for turning tapered and contoured surfaces to a template and for recessing.
2. In respect to *arrangement* as central (along the spindle axis) for drilling, boring, thread-cutting and straight-turning operations, side for end-working and cross slides in the horizontal plane, or vertical, inclined or fan-like (Fig. 28) for the vertical tool slides (square to the spindle axis).
3. In respect to the *kinematics of their motion* as slides with rectilinear travel (the most common case), and rocking slides (seldom used, see Fig. 29).

Each tool slide has its independent feed from a special cam.

Positive stops are used to raise the machining accuracy. They enable all backlash to be eliminated in the elements of the slide drive, ensuring operation with a definite stable slide deflection value and thereby raising the accuracy of performance. The same purpose is served by micrometric screws used to adjust the cutting tools.

In turning with a form tool held in a horizontal cross slide, the cutting force has a tendency to pull the slide up away from the guides or ways.

The cutting force applies a load to cross slides arranged in a vertical plane. It acts parallel to the plane of the guides and square to the direction of slide travel. Neither case favours the operation of dovetail guides, for which reason

flat ways or guides are widely used for tool slides. Dovetail guides are used only for light loads or for crowded conditions as in the fan-like arrangement of vertical slides.

Devices for retracting the tool from the work during slide return (tool relief) are not used in single-spindle automatic screw machines. Tool relief is obtained in these automatics as a result of deflection of the tool and its holder under load with the tool nose set slightly above the line of centres in external turning operations.

Tool Slides of Single-Spindle Semiautomatic Lathes

These tool slides are classified according to the following features:

1. In respect to the *direction of travel* in reference to the spindle axis as cross slides, longitudinal slides (for central and boring operations), longitudinal carriages with cross slides for withdrawal, approach, feed-in and tool relief (see Figs. 53 and 54); longitudinal carriages with a tracer-controlled cross slide for withdrawal, approach, feed-in and cross feed of the tool (semiautomatic tracer-controlled lathes) and longitudinal carriages with angular tracer-controlled slides for withdrawal, approach, feed-in and cross feed of the tool (semiautomatic tracer-controlled lathes without automatic control of the longitudinal carriage feed).
2. In respect to the *kinematics of their motion* as cross slides with a rocking motion and slides with a rectilinear motion.
3. In respect to *arrangement* see Section 6-4 since slide arrangement is closely associated with the layout of these machines.

Cross slides of the *rocking type* are simple in construction. If they are cam-actuated, they offer wide opportunities for varying the tool motion cycle (see Fig. 33), imparting versatility and processing flexibility to the lathe.

The drawbacks of these slides are their insufficient rigidity (they are mounted on long bars), and the difficulty in disposing of the continuous chips formed in the operation of carbide-tipped tools. These drawbacks restrict the use of such tools in semiautomatics with rocking slides.

The change in the cutting angle of the tool as it approaches the spindle axis degrades the performance of form tools mounted in the rear cross slide or in the front slide for tracer-controlled turning. It becomes necessary to make suitable corrections in the template profile or to provide a small tracer-controlled slide on the front slide in place of the toolholder. The lack of space does not allow such a slide to be designed with sufficient rigidity for satisfactory operation. These shortcomings of rocking tool slides have prevented them from being applied to any appreciable extent in single-spindle semiautomatics.

Slides with a straight-line motion, notwithstanding their more complex construction, are the main type used in these lathes.

The available space in a single-spindle automatic enables the guides or ways of the slides to be designed with a more favourable shape for withstanding the effective cutting forces (see Fig. 53).

Tool relief is obtained in many models during rapid return of the slide (or carriage) by the relative shift of two bars with bevelled recesses and projections (see Fig. 54).

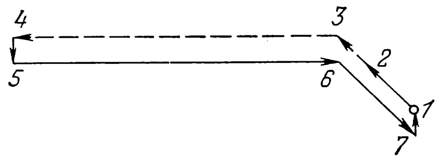


Fig. 114. Working cycle of tool motions for the front (longitudinal) carriage of a multiple-tool lathe:

1-2—angular approach; *2-3*—angular feed-in; *2-3-4*—working feed; *4-5*—tool relief; *5-6-7*—rapid return; *7-1*—tool advance

Cam bars are extensively employed for transverse withdrawal of the tool to remove the finished work and for subsequent approach and feed-in, as in Fig. 53. In this case, the movable nut carrying the roll which follows the cam bar operates in the same manner as a tracer-controlled upper (or cross) slide. Consequently there is only a single cross slide which serves both for withdrawal and for setting the tool to the required depth of cut.

Similar arrangements are used for the retraction, withdrawal and approach of cross slides in the longitudinal carriages of multiple-tool semiautomatics in which the carriages are hydraulically actuated. The working cycle of tool motions for a front (longitudinal) carriage is shown in Fig. 114.

Because of the adverse cutting conditions, the cross slides of semiautomatic lathes tend to vibrate. In certain models, motion is transmitted to the cross slide through a wedge-shaped cam linked to the rod of a longitudinal cylinder (Fig. 115). This construction does not increase the stability but enables the size of the slide to be reduced and several slides to be powered by a single cylinder.

The cam actuates shoe *1* secured to ram *3* and thus traverses cross slide *4*, to which the ram is fastened by screws and T-strip *6*. The shoe is kept in contact with cam *2* by spring *7* acting through housing *8* which is secured to ram *3*. After loosening screw *5*, screw *9* can be rotated through bevel gearing by turning a crank handle put on a square shank on top of the slide. Since the nut of screw *9* is secured to the ram, rotation of this screw is used to adjust the position of the cross slide in respect to the blank which is located above the slide.

End Tool Slides of Multiple-Spindle Automatic Bar and Semiautomatic Chucking Machines

End, or main, tool slides may be of the single-position (with a tool slide in each position) or multiple-position type (in which one tool slide serves all the positions).

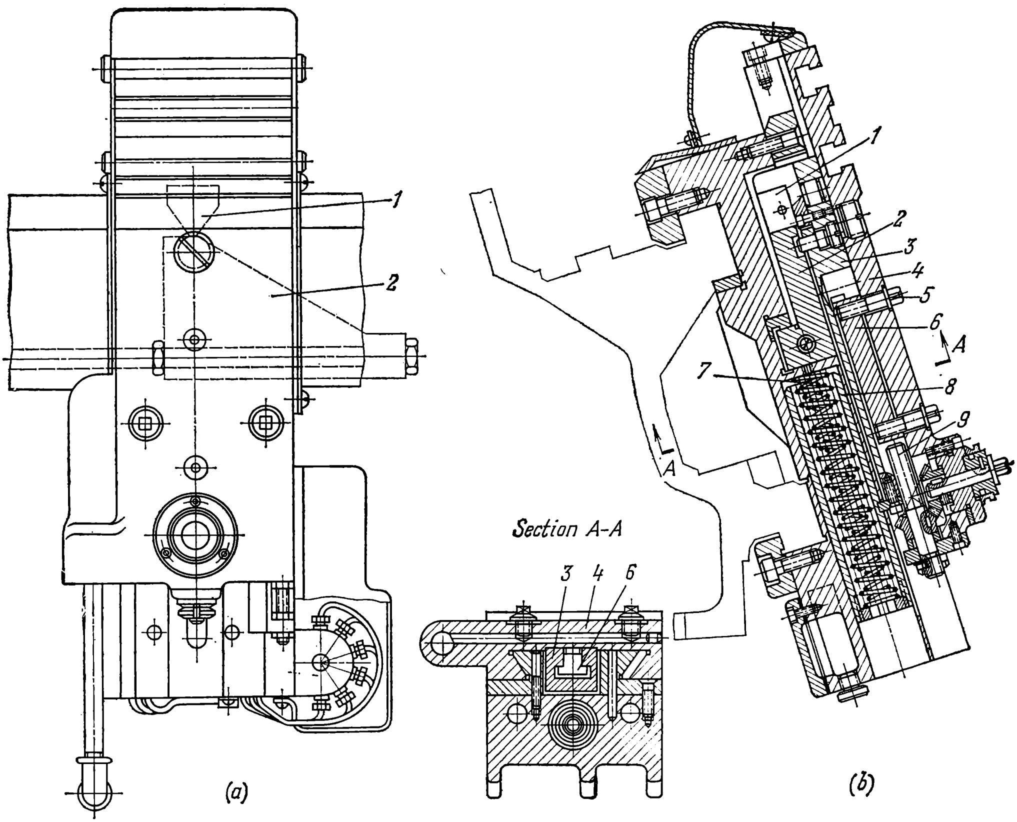


Fig. 115. Facing (cross) slide of the model 1712 tracer-controlled semiautomatic lathe

Central end tool slides of the multiple-position type are used almost exclusively in all up-to-date bar and chucking machines. They are mainly guided in travel by the carrier stem, one end of which is fixed rigidly in the spindle carrier. An additional guide is usually provided above on the top brace through which the camshaft passes.

The advantageous features of this design are the high rigidity of the tool slide, strict alignment of the end tool slide with the spindle carrier even after wear of the carrier seat in the headstock, protection of the guiding surface (carrier stem) against chips, convenient chip disposal and the symmetrical shape of the tool slide which, in conjunction with the coaxiality with the carrier, enables the tooling to be unified so that the toolholders are interchangeable.

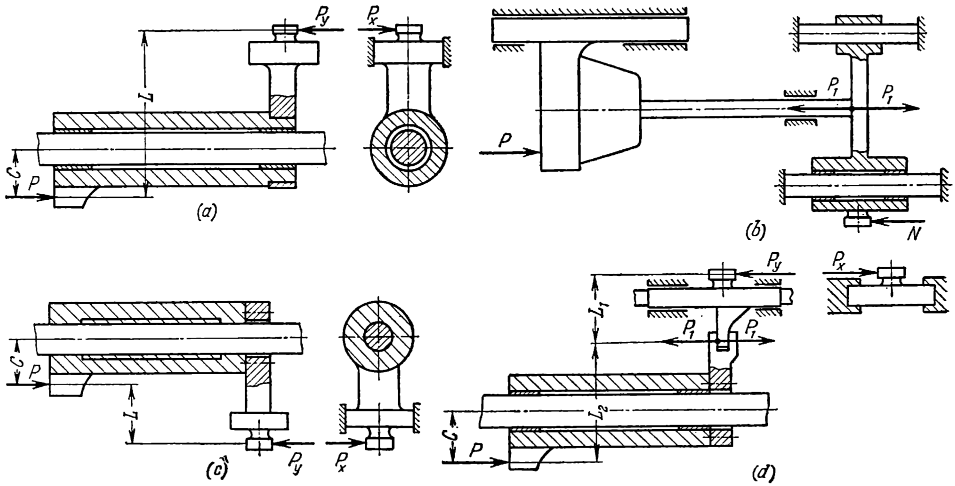


Fig. 116. Diagrams showing the application of load to the end tool slide:
(*a*) in the Conomatic (USA); (*b*) in the Gildemeister (FRG); (*c*) in model 1261 (USSR); (*d*) in models ГАШ-6 and 1240 (USSR)

The difficulty of setting up tools located in the lower positions of the end tool slide is its main drawback.

Cutting thread (in the last upper position) and drilling (in the next to the last and preceding side positions), require additional rotation of the auxiliary tool spindles running in holders mounted on the end tool slide and independent feed with rapid approach and withdrawal from positive-return cams (see Fig. 21). In the lower positions, facilities are provided only for additional rotation of the tool spindles without the independent feed feature.

The maximum load due to the cutting force is in the two lower spindle positions. If the camshaft is located above in the top brace, the distance L is comparatively large between the lines of action of feed force P_y and cutting force P.

The moment on the end tool slide (Fig. 116) from this pair of forces leads to reactions on the guide bushings and to forces which increase the equivalent coefficient of friction in the follower of the cam mechanism (see p. 31) and thereby limit the slope angle of the end tool slide cam. If the follower roll, cam and camshaft are arranged below the tool slide, the moment is reduced (see Fig. 116*b* and *c*) but the drive arrangement hinders chip disposal.

The introduction of a supplementary slide, operating with a smaller arm (L_1), improves the performance of the cam mechanism.

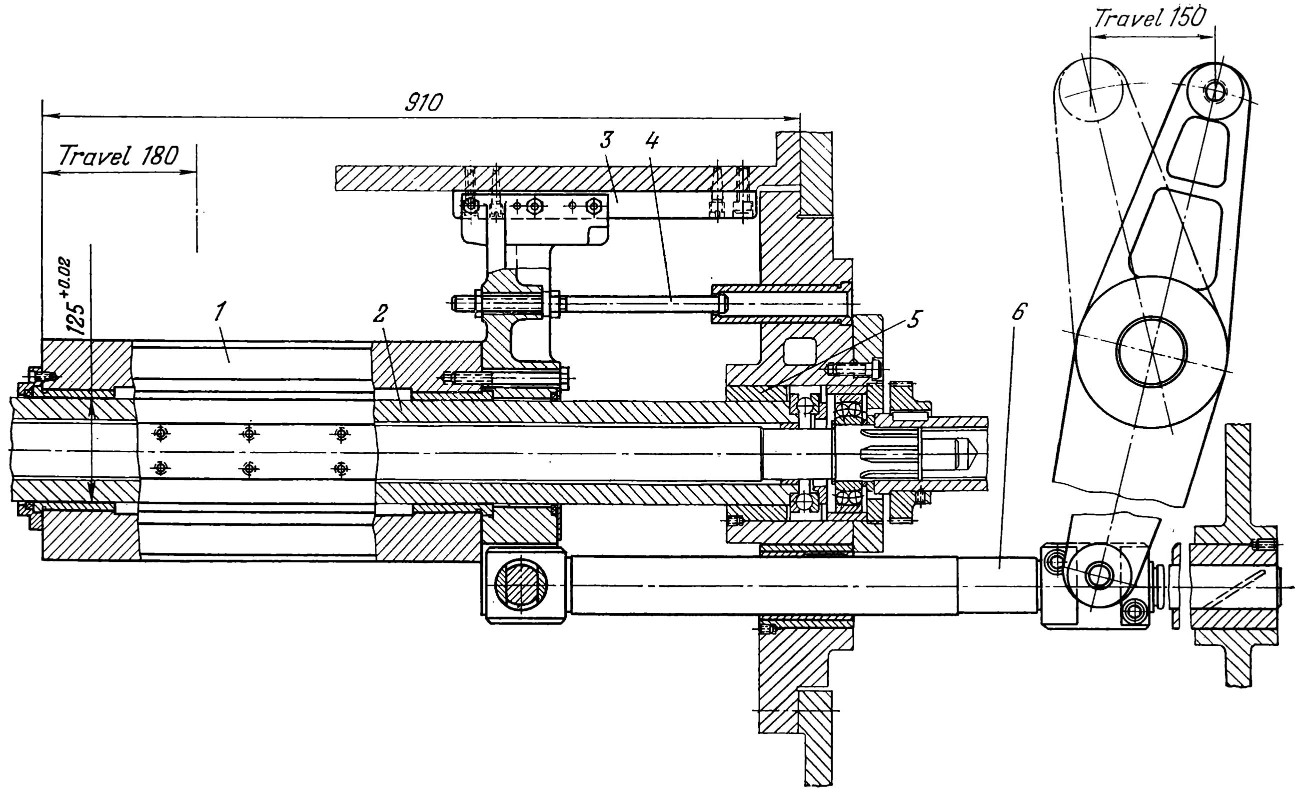


Fig. 117. End tool slide of the model 1265 automatic:

1—tool slide; *2*—carrier stem; *3*—upper guide; *4*—tie-rod and stop of the tool slide; *5*—plain bearing and ball thrust bearing of the carrier stem; *6*—connection for traversing the tool slide

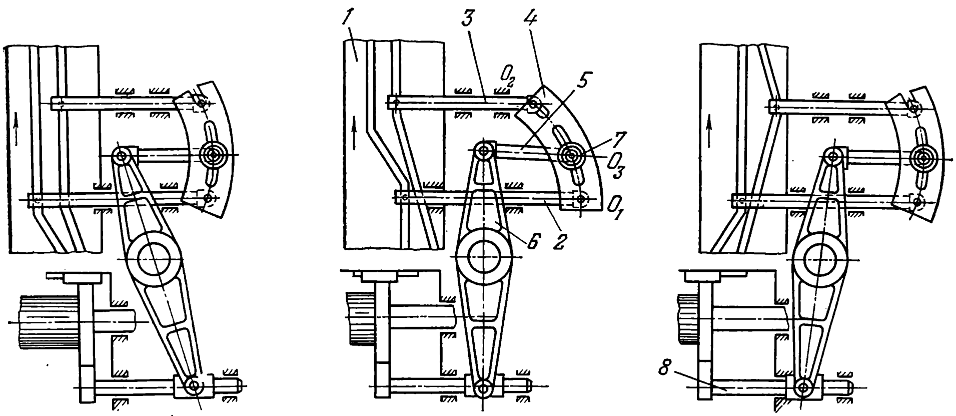


Fig. 118. Operation of the end tool slide drive in the model 1265 automatic

A more efficient solution can be found in the construction of the drive for the end tool slide of the model 1A225 automatic (see Fig. 22). The application of a lever system in conjunction with double-disk positive-return cams arranged crosswise and mounted on a branch of the camshaft enabled the arm L of the feed force to be substantially reduced, friction to be reduced by eliminating the slide block, and better access to the tool slide to be provided.

The arm length has been reduced to the minimum in the end tool slide drive of the model 1265 automatic (Fig. 117). In this drive (Fig. 118), the tool slide is actuated by positive-return cam drum *1*, mounted on the upper longitudinal camshaft, through sliding members *2* and *3*, lever *4*, connection *5*, lever *6* and tie-rod *8*.

The tool slide is set up to the required length of working travel by adjusting pin *7* along the slot of lever *4*. This lever pivots about axis O_1 or O_2, depending upon the motion of sliding members *3* and *2*.

Side Slides of Multiple-Spindle Automatics

Four-spindle automatics have four side (cross) slides; six-spindle automatics have four, five or six; and eight-spindle automatics have five or six.

Each side slide can be actuated by a separate disk cam, the length of travel being set up by adjusting the arm length of the driving lever (Fig. 119) or intermediate lever (Fig. 120).

Distinction is made between two types of side (cross) slides.

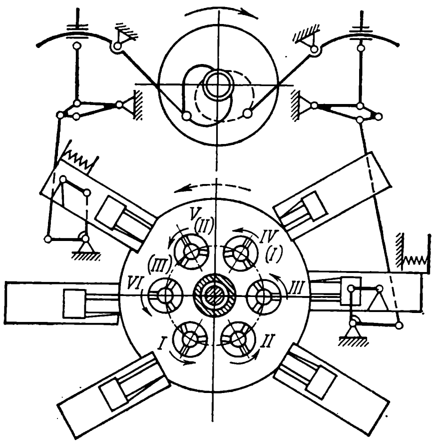


Fig. 119. Diagram of the drives of the side (cross) slides in the model 1A225 automatic

In the *slot-type cross slide*, the principal plane of the vee ways is located vertically (Fig. 121). Excess clearance is eliminated by adjusting the lower way by means of two screws *1* in pin *2*. The way is fastened with screws *3*.

As in the case of semiautomatic lathes, the cross slides of multiple-spindle automatics tend to vibrate because of the adverse cutting conditions. Hence, to ensure smooth travel, a branch of the camshaft is brought up to each lower slide, and the intermediate members (levers and tie-rods) of the drive from the disk cam to the slide are reduced to a single lever.

Chip disposal is easier in an automatic with slot-type cross slides and the chips do not foul the ways. The drive of the slides is simpler and the slides can be more easily unified in design. Though the shape of the ways is designed for an overhanging load, slot-type slides are considered insufficiently rigid for turning with a heavy chip and a large tool overhang.

In the *flat-type cross slide*, the principal plane of the ways is horizontally located. Flat ways are used. This design of slide is considered to be more rigid than the slot type. Its drawbacks are more difficult chip disposal from the lower slides and more complicated conditions for the drive.

Of especial importance to side slides is the system of positive stops whose design should compensate for the errors in the positions of the spindles in the carrier.

To avoid the extra load on the spindle carrier, the positive stops are mounted on the headstock. Each side slide has one adjustable stop for each spindle position on the carrier. Upon each indexing of the carrier, a link is shifted to bring the positive stop button in line with the adjustable stop for the next spindle position.

In another design version, stops which compensate for the error in the position of each spindle in the carrier are arranged on a disk which indexes together with the spindle carrier. In this case, only one stop is mounted on the side slide.

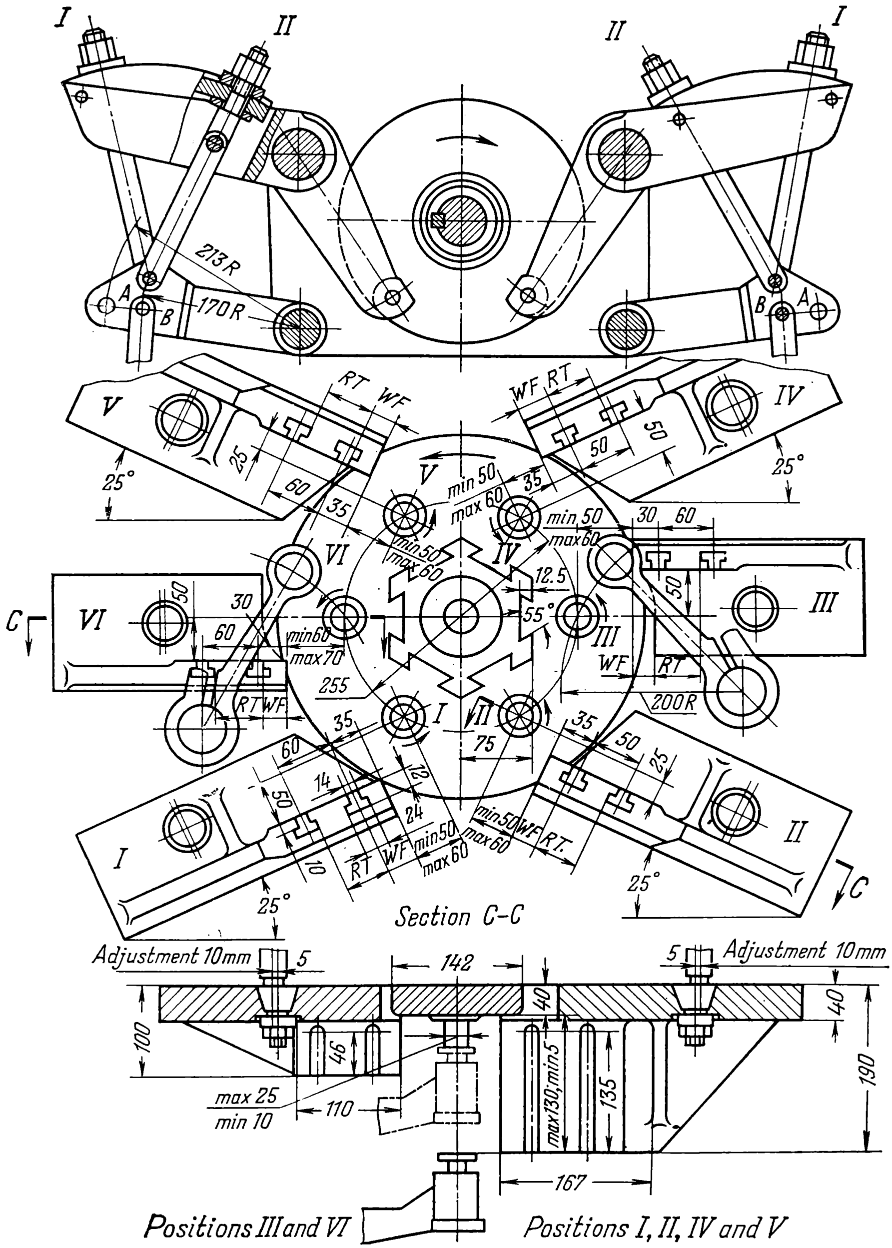


Fig. 120. Setting up the length of travel of the side slides in the model 1A225 automatic

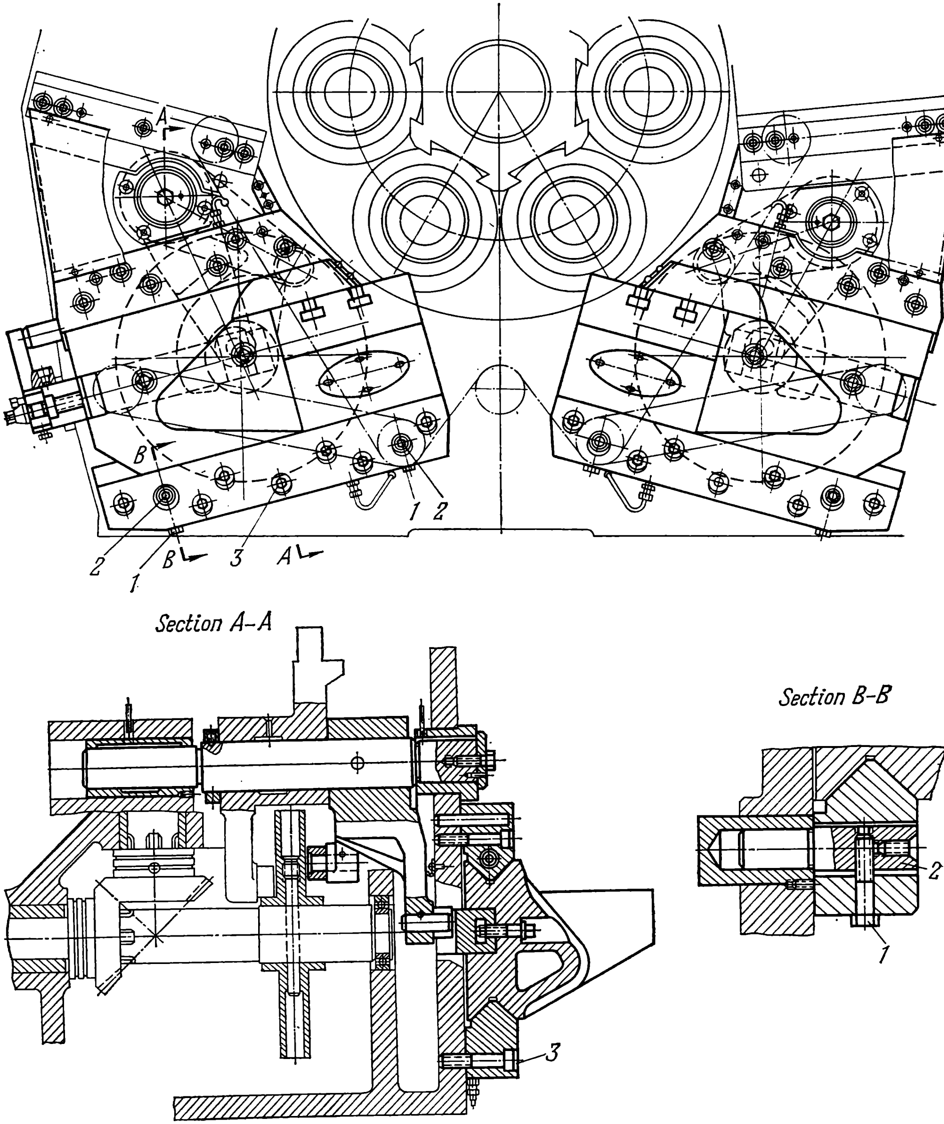


Fig. 121. Lower side slides of the model 1265-6 six-spindle automatic bar machine

CHAPTER 6

LAYOUTS OF AUTOMATIC AND SEMIAUTOMATIC MACHINE TOOLS

In designing automated machine tools much thought is given to the proper arrangement of the units, since it affects the efficiency and the field of industrial application of the machine (especially if it is to be built into an automatic transfer line). In solving this problem it is necessary to take into consideration, not only the general requirements and factors influencing the layout, but also the particular features of the drive, operative mechanisms, the construction of the working members, the purpose of the machine tool being designed and the manufacturing process being applied. Therefore, the development of the layout begins with the detailed working out of the setup and operation sequence diagrams, diagrams showing the operation of the main working members (slides) and taking into account their operative mechanisms, and diagrams showing how the blank is to be loaded and the finished workpiece removed and otherwise handled.

6-1. General Requirements Made to the Layout

The following requirements are made to the layout of a machine tool being designed:

1. The given or standard manufacturing process should be efficiently accomplished, and provision made for the required output and machining accuracy.
2. Ample rigidity and vibration-proof properties should be ensured without increasing the weight of the machine. Rigidity may be obtained by the use of a frame-type construction or box-shaped components or both.
3. The machine should be convenient to produce, both in the manufacture of the parts and in their assembly. It should be easy in maintenance and accessible in repairs.
4. Operator safety is a prime factor.
5. It should allow convenient loading of the bar stock by hand in a bar automatic or of the blank by hand or with a hoist or crane in a semiautomatic lathe.
6. It should allow various units and auxiliary devices to be built in conveniently.

7. It should occupy minimum floor space for installation and servicing.

If the machine tool is to be built into an automatic transfer machine or line, the following factors become decisive:

8. The possibility of continuous automatic chip disposal from the cutting zone and from the machine tool.

9. Proper co-ordination of the layout of the machine with the handling systems of the automatic transfer machines and with the loading devices used for blanks that are to be machined.

6-2. Single-Spindle Automatic Screw Machine Layouts

With rare exceptions, a horizontal layout is used for these automatics. The cross tool slides are arranged horizontally (front and rear slides) and vertically (up to three slides in fan-like order), while the longitudinal slides are centrally located for centred cutting tools (Figs. 122 and 123) and for a turret (Fig. 124).

The front and rear cross slides (Fig. 122) or only the front slide of turret-type automatic screw machines (Fig. 124) may be of compound design to allow for longitudinal turning behind a shoulder, taper or contour turning to a template and the recessing of holes. These automatics may have as many as five or six cross slides.

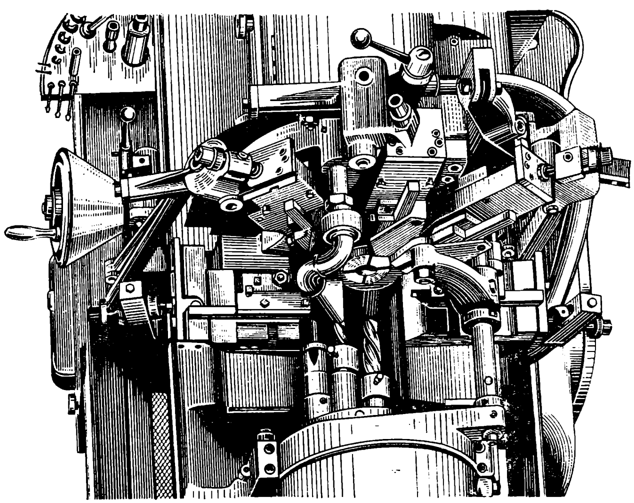


Fig. 122. Working zone of the Manurhin automatic screw machine, model PF (France)

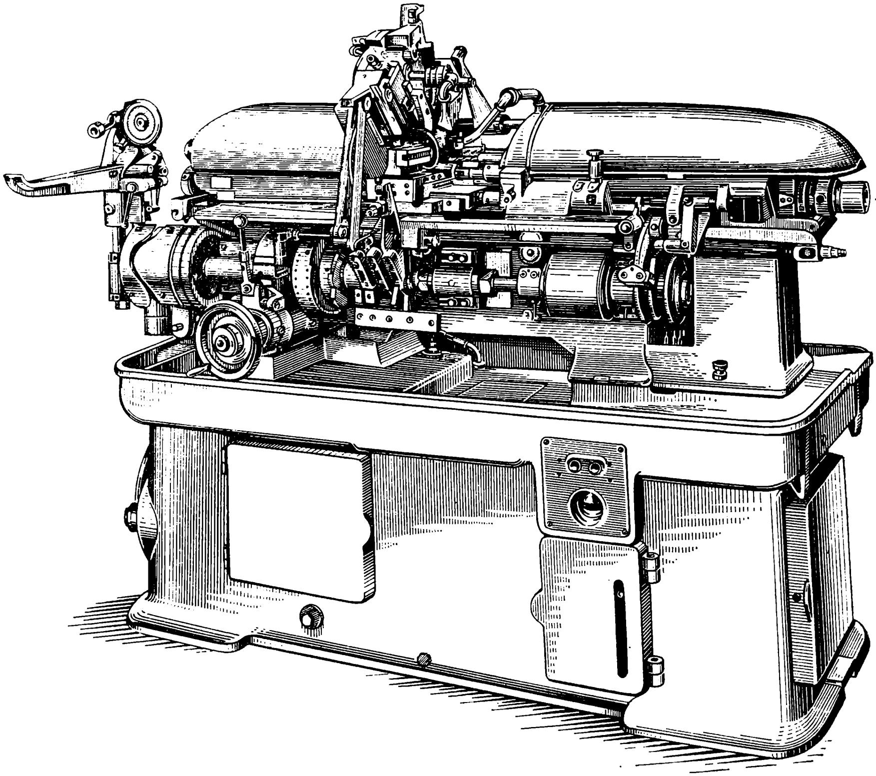


Fig. 123. Model **PF automatic screw machine (Manurhin, France)**

The fan-like arrangement of the cross slides in a vertical plane in the tool frame of a Swiss-type automatic screw machine enables the cutting tools to operate directly adjacent to the guide bushing with very short overhang of the bar. This, in conjunction with the provision of the guide bushing and longitudinal bar feed, contributes to the high machining accuracy.

The small size of the stock and the short lengths of working travel, due to the large number of slides and cutting tools, lead to the formation of short chips. On the other hand, the limitation of the cutting speed for a number of reasons (see p. 174), even when carbide-tipped tools are used, and the abundant supply of cutting fluid, favour the formation of a sheared chip, increase its brittleness and facilitate chip breaking. Hence, with a sufficiently voluminous recess in the bed and a sufficient distance between the cross slides, no difficulty is encountered in disposing of small chips from the cutting zone.

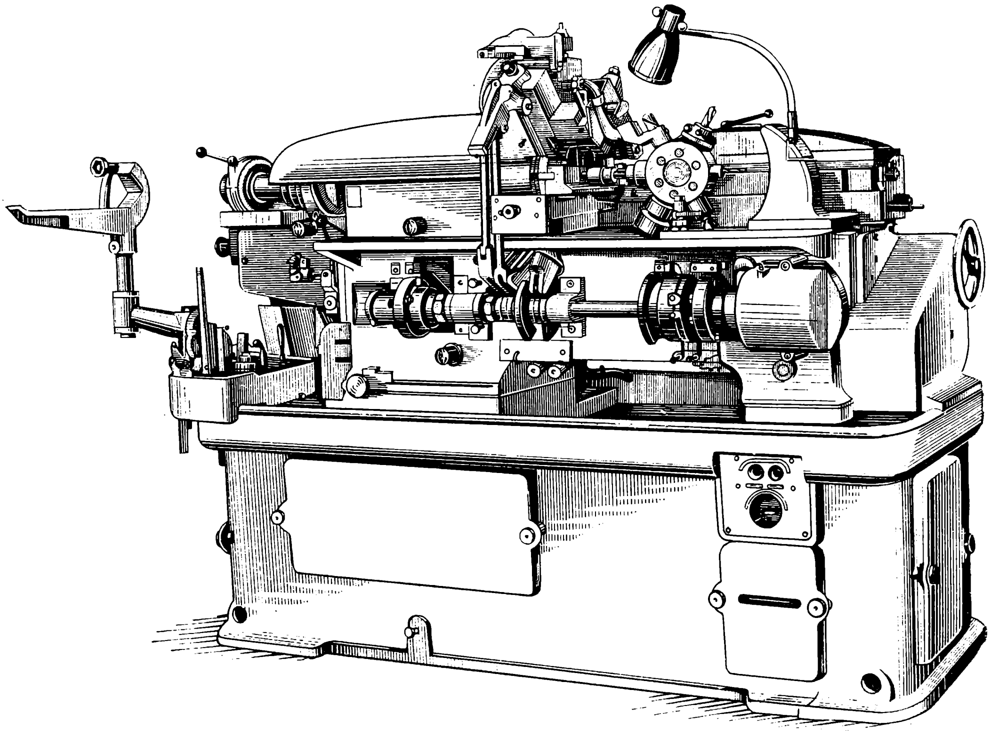


Fig. 124. Model TR turret-type automatic screw machine (Manurhin, France)

Horizontally arranged single-spindle automatic screw machines can be readily adapted to magazine and, consequently, to hopper feeding of the blanks, this being the common method used in automatic transfer machines for handling parts machined in such automatics.

6-3. Multiple-Spindle Automatic Bar Machine Layouts

The lower accuracy attainable (in comparison with single-spindle automatics) due to errors in the positioning of the spindles in a multiple-spindle carrier and to the error in the carrier itself when it is locked after indexing, has led to the use of a centrally arranged end tool slide located on a stem whose one end is fixed in the carrier.

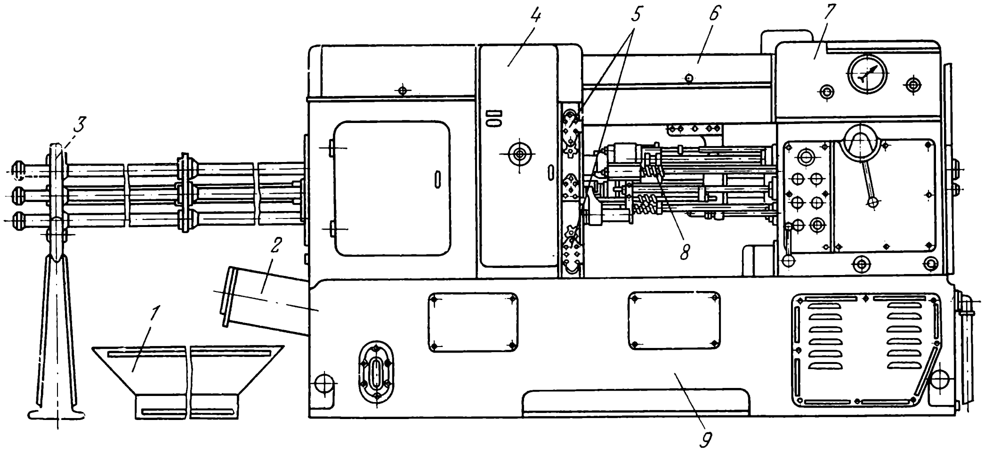


Fig. 125. Six-spindle automatic bar machine, model 1A225:
1—chip box; *2*—chip conveyer; *3*—stock reel stand; *4*—headstock with spindle carrier; *5*—side (cross) slides; *6*—top brace with camshaft; *7*—gearbox; *8*—end tool slide and independent-feed tool spindles; *9*—base

In this layout, the base, headstock (housing the spindle carrier), gearbox and top brace (housing the camshaft) form a rigid closed frame (Fig. 125) carrying the tool slides and other mechanisms.

The upper-positioned camshaft and the central end tool slide facilitate chip disposal. Slot-type cross slides are used for the same reason.

Drawbacks of this layout are the crowded arrangement of the spindles in the carrier, drastically limiting their radial overall size, and the lack of space which leads to difficulties in setting up tools on the two lower positions of the end tool slide.

6-4. Single-Spindle Semiautomatic Lathe Layouts

The ample overall dimensions of single-spindle semiautomatics enable a rugged spindle unit of massive design to be used, together with a powerful drive. This also concerns the carriages and slides in general-purpose models which are consequently capable of performing roughing operations with carbide-tipped tools at the highest cutting speeds permitted by tool life considerations. With the extensive application of hydraulic tracer-controlled slides in general-purpose semiautomatics and single-tool setups instead of the multiple-tool type, the cutting speed can be increased substantially.

All of these circumstances lead to the formation of a great amount of continuous chips. In tracer-controlled turning, chipbreaker operation becomes

unreliable because of the variable cutting speed and chip cross section. Under such conditions, dependable automatic continuous chip disposal from the cutting zone is one of the main and most difficult problems in machine layout. Unless this problem is solved, the lathe cannot be built into an automatic transfer machine or line.

Another problem, of no less importance, is to co-ordinate the layout of the single-spindle semiautomatic with the handling and loading systems of the automatic transfer machine. Possibilities of such co-ordination should be wider in the general-purpose models, enabling them to be coupled in a transfer line with machine tools of other purposes and layouts, requiring various types of loading and handling systems.

Horizontal layouts have the front and rear carriages arranged horizontally or at an angle (up to 30°) to the horizontal. They have a bed in the form of two box-shaped beams with a wide opening between them over the full length of the cutting zone. The longitudinal and cross carriages (or slides) are mounted on the front and rear beams, without covering the opening between them.

Such a layout is convenient for servicing the machine (all units are accessible to the operator from floor level) but also has certain shortcomings:

1. The layout is not of the closed type and the rigidity of the bed is lower than it could be with a frame-type layout.

2. The operation of the tools on the rear slide is poorly visible from the operator's workplace.

3. Accessibility to the rear slide (or carriage) is difficult from the operator's workplace.

4. Chip disposal is improved in comparison with general-purpose engine lathes, in which the opening in the bed is not continuous, being covered in part by the saddle of the carriage, but here also the carriage units and tooling may prevent continuous chips from dropping freely through the bed. Uninterrupted disposal of continuous chips cannot be ensured in semiautomatic lathes with a horizontal layout.

5. Blanks can be conveniently loaded only from above, dictating an overhead arrangement of the handling line in the automatic transfer machine. If this handling line is arranged at the side, passing along in front of the lathe, provision must be made for overhead transfer of the blank from the handling line to the loading position. Through-horizontal blank transfer, with the handling line passing through the clamping position (the between-centres part of the lathe axis), is impeded by the carriage slides.

Frame-type (*portal*) *layouts* with the longitudinal carriages arranged above on a sort of crossrail have been used in various versions, all providing for a satisfactory disposal of continuous chips.

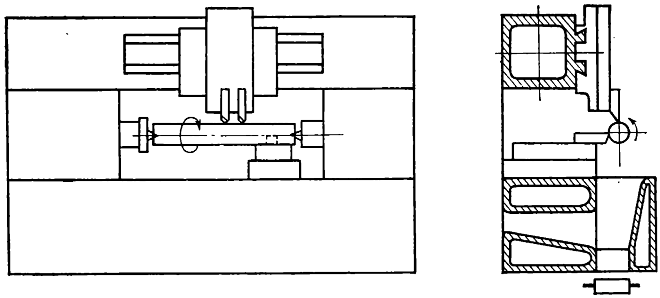


Fig. 126. Frame-type layout with vertical arrangement of the longitudinal carriage on the crossrail

In a layout with the longitudinal carriage arranged vertically (as in the Hasse & Wrede semiautomatic lathe for turning medium-size shells, whose layout is shown schematically in Fig. 126), the front carriage is more easily accessible for setting up tools, and blanks can be conveniently loaded and workpieces unloaded by hand. The top carriage, however, interferes with crane loading from above. Moreover, it is difficult to set up the tooling in the rear cross slide.

If the longitudinal carriages are arranged horizontally underneath the crossrail (Fig. 127), as in the model 1832 (K175) semiautomatic lathe for roughing the axles of steam locomotives, both vertical and horizontal modes of loading of the blank are possible, but the visibility of the cutting zone is poor. Accessibility to the longitudinal carriage is inconvenient. This has been taken into account and alleviated to some extent in the design of the toolholder (Fig. 128). Conditions for continuous chip disposal have been considerably improved in this layout, an open free space being provided for the dropping of the chips downward, in which direction they are diverted by the front end of the toolholder. In respect to the conditions of chip disposal from the longitudinal carriage, this is one of the most expedient layouts of semiautomatic lathes. The designers of the Krasny Proletary Plant rejected the horizontal layout for the locomotive axle lathe after investigating the possibilities of a frame-type layout.

Frame-type layout with the carriages inclined at an angle of 15° to the vertical. The multiple-tool semiautomatic lathe, model 1721, and the hydraulic tracer-controlled semiautomatics, models 1722 and 1712 (Fig. 129), are examples of this type of layout. The distinguishing feature of this layout is the arrangement of the bed ways for the longitudinal (top) carriage and the cross (lower) carriage in a plane 15° to the vertical (Fig. 130). Its advantages are:

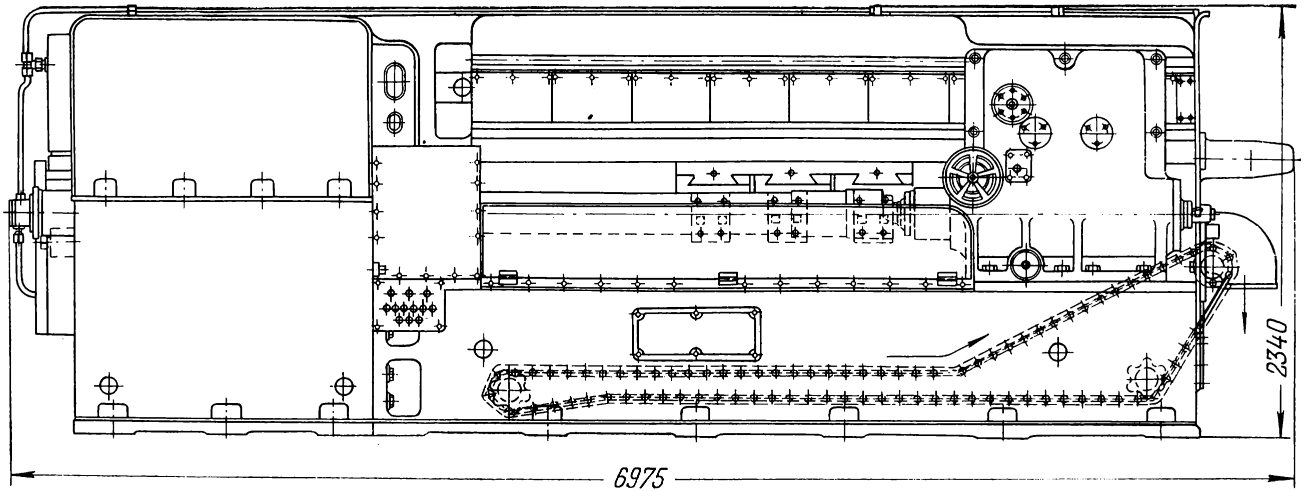
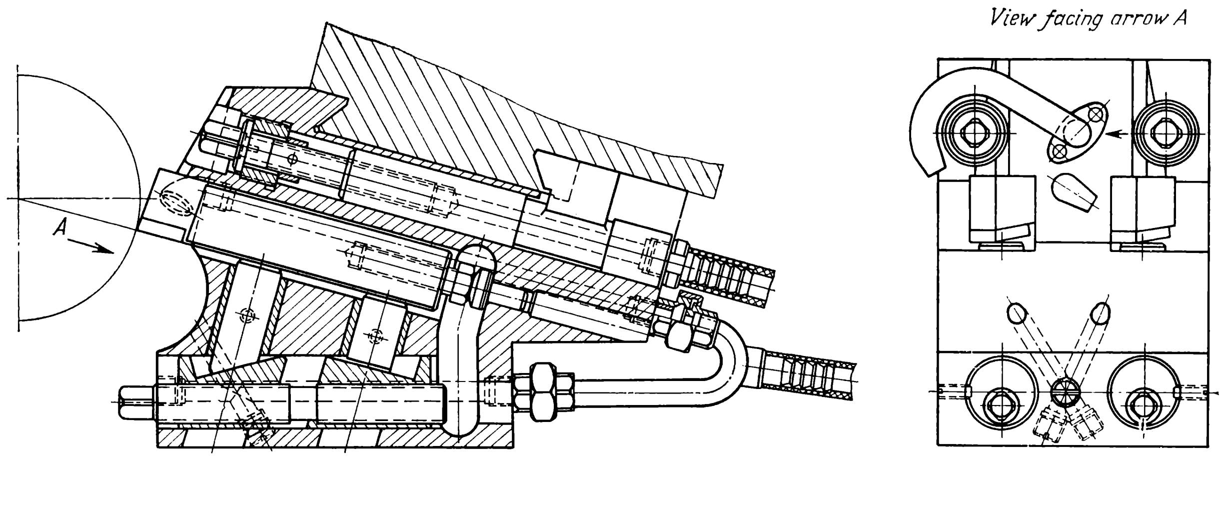


Fig. 127. Semiautomatic multiple-tool lathe, model 1832 (K175), for roughing locomotive axles (Krasny Proletary Plant, Moscow)



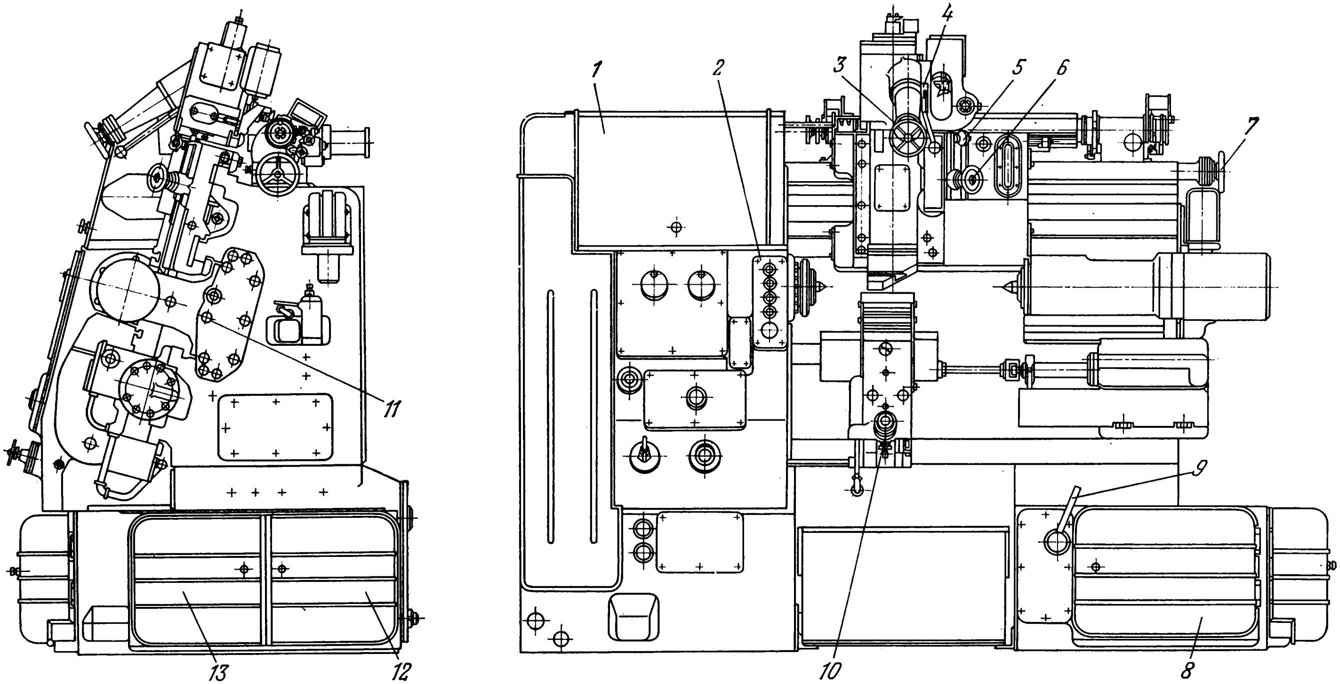


Fig. 129. Semiautomatic tracer-controlled lathe, model 1712:

1 and *2*—setting-up and operative control panels, respectively; *3*—handwheel for setting the tool to the depth of cut; *4*—lever for clamping the stylus slide; *5*—knob for clamping the stylus stop; *6*—handwheel for adjusting the stylus stop; *7*—handwheel for axial setting of the template; *8*—control panel for the tracer-controlled slide; *9*—tailstock spindle control lever; *10*—square shank for setting the tool to the depth of cut; *11*—screw for traversing the tailstock; *12*—control panel for the tailstock spindle; *13*—control panel for the cross slides

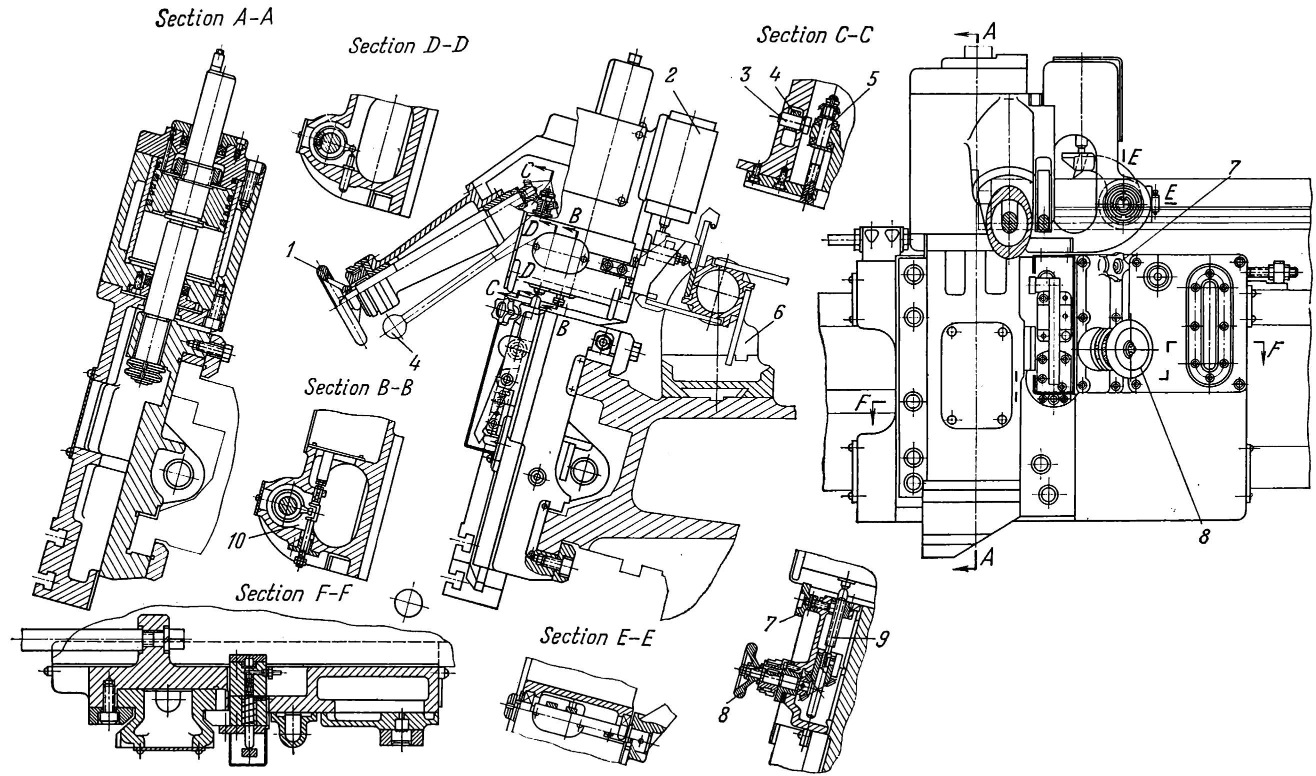


Fig. 130. Tracer-controlled slide of the model 1712 semiautomatic lathe:

1—handwheel for turning screw *5*; *2*—stylus valve body; *3*—eccentric for clamping the stylus valve slide; *4*—lever for turning eccentric *3*; *5*—screw for adjusting the stylus valve slide and stylus tip in setting the tool to the depth of cut; *6*—template drum; *7*—knob for clamping stylus stop *9*; *8*—handwheel for adjusting the stylus stop; *9*—stylus stop; *10*—mechanism for shifting the stylus tip and stylus valve spool with solenoid Sd_5 (see the hydraulic circuit diagram in Fig. 66)

(1) near approach to the blank and tools from the operator's position for manual or crane loading, setting up and adjusting the tools and for measuring the workpiece;

(2) provision can be made for a convenient shield;

(3) the operation of the tools is visible from the operator's position;

(4) convenient chip disposal, though it is hindered to some extent by the cross (front) slides;

(5) exceptional versatility in respect to co-ordination with the handling system of automatic transfer machines or lines, as the blank can be loaded in a horizontal plane from the front of the lathe if the handling line is arranged laterally along the front of the component machine tools or if through-horizontal blank transfer is used with the handling line passing through the clamping position (see the front view in Fig. 129 in which the slides are withdrawn when the blank is being transferred). Top loading is also possible.

Vertical layouts. Vertical single-spindle semiautomatic lathes are available in two design versions: with the spindle above, which is convenient as to chip disposal in turning internal surfaces, and with the spindle below. The second arrangement is more widely applied. This layout has the following advantages:

(1) good visibility and accessibility of the cutting zone and slides from the operator's position;

(2) high design rigidity;

(3) reduction of the occupied floor space by about 40 per cent as compared to a horizontal layout;

(4) satisfactory chip disposal.

Drawbacks of this layout are the limited length of workpiece that can be handled, the large weight of the lathe, and the limited loading possibilities in an automatic transfer line. Only side loading can be used with the handling line arranged laterally along the front of the machine tools.

6-5. Multiple-Spindle Semiautomatic Chucking Machine Layouts

Blanks up to 200 mm in diameter are machined in progressive-action semiautomatic chucking machines with a horizontal layout, similar to that of the automatic bar machines with which they are widely unified.

Vertical layouts are used for progressive-action chucking machines for turning blanks 200, 300, 600 and 800 mm in diameter. The advantages of vertical layouts include the following:

(1) wear of the carrier bearings is more uniform and its effect on machining accuracy is less;

(2) space conditions are more favourable for the spindle units and their drives;

(3) the spindle units are of highly rigid design;

(4) it is possible to provide for different spindle speeds and slide feeds at each station;

(5) the ample space conditions enable as many as 16 spindles to be used;

(6) the general rigidity of the whole machine is very high;

(7) lateral hand loading and crane loading are convenient.

The conditions imposed by this layout are unfavourable for the design of the longitudinal and cross slides. It becomes necessary to mount cross slides on an additional column which cannot usually be made sufficiently rigid, or on an attached upright which increases the required floor space.

CHAPTER 7
PRODUCTION CAPACITY OF AUTOMATIC MACHINE TOOLS

7-1. Production Capacity Indices

The main index of production capacity in automatic and semiautomatic machine tools is the piece output; it is more compatible with the type and size of the blanks or stock and the cyclic operation than other indices (absolute production capacity, cutting capacity or formative capacity, see p. 14, Vol. 3).

The piece output is determined by the formula

$$Q = \frac{3{,}600}{t_{cal}} \text{ pcs per hour} \tag{31}$$

where t_{cal} is the calculated time, in sec, for machining the workpiece and is determined, in turn, from the formula

$$t_{cal} = t_p + \frac{t_{su}}{n} \text{ sec} \tag{32}$$

where t_p = standard time (time per piece), sec

t_{su} = setup time, in sec, for setting up the machine and for demounting the tooling after machining one lot

n = number of workpieces in one lot.

The standard time (per piece) is found by the formula

$$t_p = t_m + t_h + t_s + t_o + t_f \text{ sec} \tag{33}$$

where t_m = machining (or, in the general case, processing) time, sec

t_h = handling time, sec, i.e., time for the rapid traverse movements of the slides and the cycles of motion of the auxiliary working members

t_s = servicing time, sec, required for setting, adjustments, dressing, tool changing, chip removal, etc., referred to a single workpiece

t_o = time, sec, required for various organizational measures and delays

t_f = fatigue time, sec, required for rest and personal needs.

It is common practice, however, to deviate from the general method given above in determining the piece output of automatic machine tools. As a rule, the design piece output is found in which the noncyclic time losses are not

taken into account and which is based on the cycle time T in seconds. Thus

$$Q_d = \frac{3{,}600}{T} \text{ pcs per hour} \tag{34}$$

or

$$Q_d = \frac{60}{T} \text{ pcs per min} \tag{35}$$

Here the machining cycle time is

$$T = t_w + t_i \tag{36}$$

where t_w = total time required for all the working travel motions
t_i = total time required for all the idle travel of the main working members and the total motion cycle times of the auxiliary working members.

Noncyclic time losses are taken into consideration by means of the general operation factor η_o, and the actual piece output is determined from the formula

$$Q_{act} = Q_d \eta_o \tag{37}$$

Depending upon the complexity of the workpiece being machined the operation factor is taken as $\eta_o = 0.9$ for workpieces of simple shape; $\eta_o = 0.85$ for medium complexity and $\eta_o = 0.8$ for workpieces of complex configuration.

In automatic transfer machines, the operation factor is taken in the range from 0.65 to 0.8.

In his investigations concerning the production capacity of automatic machine tools, Prof. G. Shaumyan proceeds from the ideal processing production capacity $Q_{pr} = K$. Thus

$$Q_{pr} = K = \frac{1}{t_w} \text{ pcs per min} \tag{38}$$

where t_w is the total time, in min, required for the working travel motions, which do not overlap, during the machining cycle for one piece.

The cyclic production capacity of the machine tool is

$$Q_{dm} = \frac{1}{t_w + t_i} \text{ pcs per min} \tag{39}$$

where t_i is the total time required by the idle travel motions of the main working members which do not overlap and by the cycles of motion of the auxiliary members.

Substituting $t_w = \frac{1}{K}$ into equation (39) we obtain

$$Q_{dm} = \frac{1}{t_w + t_i} = \frac{1}{\frac{1}{K} + t_i} = \frac{K}{Kt_i + 1} = K\eta \tag{40}$$

and

$$\eta=\frac{1}{Kt_i+1}=\frac{Q_{dm}}{K}=\frac{t_w}{t_w+t_i}=\frac{t_w}{T} \tag{41}$$

where η = output factor of the machine tool
T = machining cycle time from equation (36).

Academician V. Dikushin calls the ratio $\frac{t_w}{T}$ the degree of continuity of the manufacturing process. It is equal to the output factor of the machine tool proposed by Prof. G. Shaumyan.

It can be seen from the expression $\eta = \frac{1}{Kt_i+1}$ that at $t_i = \text{const}$, η is reduced with an increase in K, the ideal processing capacity. In order to obtain an increase in η with K, it is necessary, at the same time, to reduce t_i. This means that along with improvements in the manufacturing process, it is necessary to improve the design of the machine as well. If this condition is complied with, then, as can be seen in equation (40),

$$Q_{dm}=\frac{1}{\frac{1}{K}+t_i} \quad \text{at } K\to\infty \text{ and } t_i\to 0$$

there is no limit theoretically to the increase in production capacity of the machine tool.

Automatic and semiautomatic lathes have the following output factors:

for single-spindle automatic screw machines, $\eta = 0.35$ to 0.45;

for horizontal multiple-spindle automatic bar and semiautomatic chucking machines $\eta = 0.65$ to 0.85;

for turret-type automatic screw machines, $\eta = 0.50$ to 0.60;

for semiautomatic multiple-spindle vertical chucking machines $\eta = 0.75$ to 0.85.

The actual output is determined by taking into account the noncyclic time losses t_l. Hence

$$Q_{act}=\frac{1}{t_w+t_i+t_l}=\frac{1}{\frac{1}{K}+t_i+t_l}=\frac{K}{K(t_i+t_l)+1}=K\eta_g \tag{42}$$

where the general output factor is

$$\eta_g=\frac{1}{K(t_i+t_l)+1} \tag{43}$$

7-2. Methods of Increasing the Production Capacity

To increase the production capacity of an automatic machine tool it is necessary to reduce the time required by the operating cycle of the machine, as well as by the elements of the calculated piece time that make up the noncyclic time losses.

The following methods are used to reduce t_m—the machining time—or, in other words, the total time t_w required for the working travel motions:

1. Concentrating cutting tools at each position or station. The degree of concentration is assessed by the concentration factor $c = \frac{m}{n}$ where m is the total number of cutting tools and n is the number of positions.
2. Overlapping working travel motions in time.
3. Providing independent spindle speeds and rates of feed at each position or station. This feature is available, for example, in the model 1A136 automatic screw machine and in the semiautomatic vertical multiple-spindle chucking machines.
4. Machining several workpieces simultaneously (in parallel).
5. Employing carbide-tipped cutting tools to enable high-velocity machining to be applied. The cutting speed in such cases may be limited by (a) insufficient rigidity of the spindles and their bearings due to the lack of room in the radial direction and to the great number of components crowded into the spindle units (or spindle carriers of multiple-spindle automatics); (b) the necessity for carrying out longitudinal turning, turning with form tools and cutting off at a single speed of spindle rotation; and (c) difficulties encountered in breaking and disposing of the chips.

The most favourable conditions for designing spindles, in respect to their radial overall size and facilities for setting up the rotational speeds, are found in single-spindle tracer-controlled semiautomatic lathes and in multiple-spindle vertical semiautomatic chucking machines.

Among the methods used to reduce the time t_i lost in the idle travel of the slides and of other main working members are the following:

1. Overlapping the idle travel motions in time with each other and with the working travel motions.
2. Reducing the number of idle travel motions by reducing the number of operation elements. This is accomplished by applying multiple-tool machining and tracer-controlled turning.
3. Increasing the speed of the idle travel motions by the provision of facilities for rapid rotation of the camshaft, special devices for rapid approach and withdrawal.

Measures that serve to reduce the time lost on the motion cycles of the auxiliary working members include:

1. Overlapping the auxiliary (handling) motions in time with one another and with the idle travel motions of the slides (indexing the turret and withdrawing the turret slide along a drop curve of the lead cam, see also the timing chart in Fig. 37).
2. Increasing the speed of the auxiliary motions (designing the drive of the operative mechanisms from high-speed auxiliary shafts or individual electric motors).

3. Automation of the handling motions in semiautomatics (loading and clamping devices).

Many methods have been devised for reducing the servicing time t_s. These include:

1. Scheduled readjustments of the tooling, in which the readjustments of the tools in a group are made to overlap in time and may be prepared beforehand.

2. Reducing the time required to readjust the tooling by using interchangeable toolholders which are set up to locating datum surfaces on the tool slide or carriage. The cutting tools are set up in the holder outside of the machine with the aid of a dial indicator. Locking member *2* (Fig. 131*a*), secured to plunger *4*, enters an angular transverse slot in the base of the tool and, under the action of spring *1*, holds the tool tightly against the locating surface in the toolholder slot and against locating screw *3*. In operation the tool is firmly clamped in the holder or tool block by the cutting force. Length settings of the tool are made in a special fixture (Fig. 131*b*) to a master tool (both the tool and master being set into body *1* of the fixture up against stops *2*) by means of dial indicators *4*. Pins *3* bear against the contact points of the dial indicators and the adjusting screws of the tool and the master.

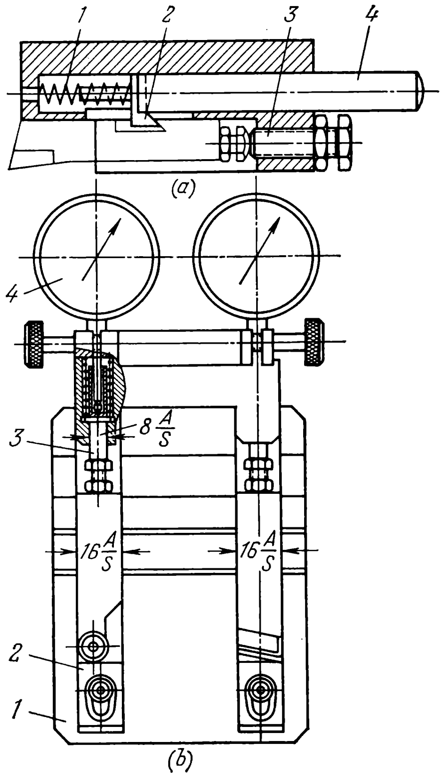


Fig. 131. Quick-change preset tool: (*a*) clamping of the tool in the holder; (*b*) fixture for making length settings of the tool outside of the machine

3. Applying single-tool machining—tracer-controlled turning.

4. Applying automatic resetting or adjustment by feedback controls.

The time t_0 for various organizational measures and delays can be reduced by improving the general production management in the plant and by employing scheduled readjustments which reduce delays in waiting for a setter-up.

The share of the setup time t_{su} included in the calculated time t_{cal} per piece can be reduced, together with the corresponding noncyclic time losses, by resorting to one or more of the following procedures:

1. Increasing the lot size, i.e., number of workpieces n in one lot.
2. Simplifying change-overs of the machine to other lengths of travel of the working members by setting up the mechanism for transmitting motion from the cams to the slides without changing cams.
3. Applying the principle of group setups (see Sec. 8-6).
4. Using quick-change setting-up facilities which are set up and adjusted outside of the machine. These include quick-change cutting tools and tool blocks, cam sets and whole machine tool units (spindle heads of unit-built machine tools).

7-3. Selecting the Number of Spindles

An increase in the number of spindles, as well as an increase in the degree of automaticity of a machine tool, complicates the setting-up procedure, increases the time required for setting up, thereby increasing the time needed to prepare the machine tool for a new lot of workpieces (Fig. 132a). On the other hand, an increase in the degree of automaticity and the number of work spindles leads to an increase in the production capacity of the machine, while the total time required to manufacture a lot varies differently for machine tools of different types with the lot size (Fig. 132).

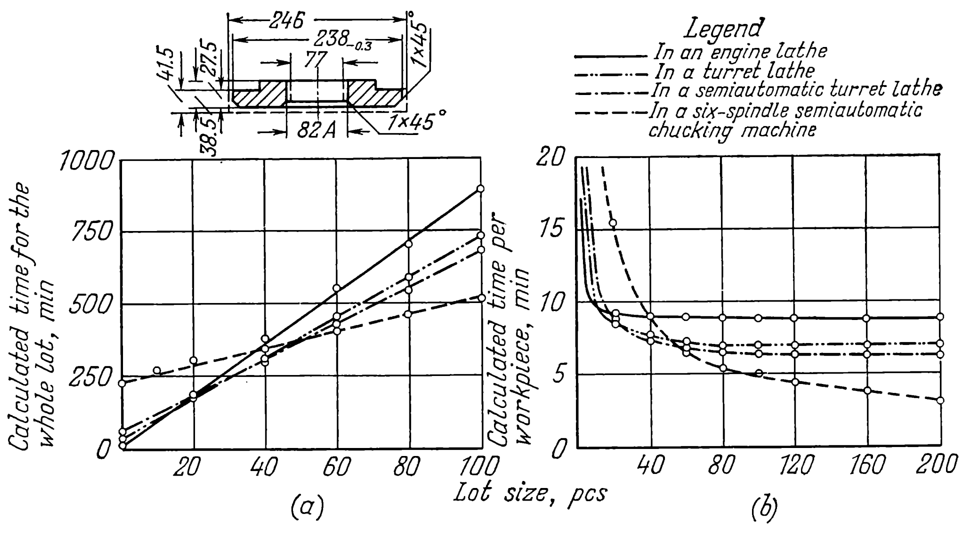


Fig. 132. Variation in the calculated time depending upon the type of lathe and the lot size:
(a) time required to machine the whole lot; (b) time required to machine one workpiece

With an increase in lot size, the time per piece varies in such a manner (Fig. 132) that each type of machine has its most advantageous range of the lot size in which the time per piece is less for the given type of machine than for machines of other types. Thus, the optimum number of work spindles depends upon the lot size.

The number of spindles is also associated with the manufacturing process; a greater number of spindles and positions or stations are required to obtain a part of complex configuration with an involved manufacturing process. An increase in the number of spindles and positions enables operation elements requiring more machining time to be broken down so as to level off the times of the component operation elements and reduce the machining time for the whole operation.

The selection of the number of spindles for an automated machine tool of interlinked design, in which the failure of one spindle leads to idle time of all the others, depends upon the relative down time of the machine due to troubles in the spindles and the corresponding tool slides.

Let us denote by:

R_1 = the down time of a spindle of the machine tool in minutes, due to trouble associated with this spindle, during a period of T minutes (for example, one shift)

$r_1 = \frac{R_1}{T}$ = relative down time of the spindle due to its own malfunctioning

$p_n = r_1 n$ = relative down time of the spindle due to its own malfunctioning and to that of the other spindles, the total number of spindles being equal to n

q_n = relative time of trouble-free spindle operation, taking into account its down time due to its own malfunctioning and to that of the other spindles.

Obviously

$$q_n = 1 - p_n = 1 - r_1 n \tag{44}$$

The total time A_n of trouble-free operation of all n spindles of a machine tool during the period T is

$$A_n = q_n nT = (1 - r_1 n)\, nT = (n - r_1 n^2)\, T \tag{45}$$

Next we determine at what value of n — the number of spindles—the time A_n reaches its maximum value. Thus

$$\frac{dA_n}{dn} = T\,(1 - 2r_1 n) = 0 \tag{46}$$

from which

$$n = \frac{1}{2r_1} \tag{47}$$

Since

$$\frac{d^2A_n}{dn^2} = -2r_1T < 0 \tag{48}$$

then at $n = \frac{1}{2r_1}$, the time A_n of trouble-free operation of all n spindles reaches its maximum value:

$$A_{n\,max} = (1 - r_1 n)\,nT = \left(1 - \frac{1r_1}{2r_1}\right)\frac{T}{2r_1} = \frac{T}{4r_1} \tag{49}$$

The time A_1 of trouble-free operation of a single-spindle machine tool during the same period T is

$$A_1 = (1 - r_1)\,T \tag{50}$$

The relative increase in output k_n due to an increase in the number of spindles from 1 to n is

$$k_n = \frac{A_n}{A_1} = \frac{(1 - r_1 n)\,nT}{(1 - r_1)\,T} = \frac{(1 - r_1 n)\,n}{1 - r_1} \tag{51}$$

and at $n = \frac{1}{2r_1}$ it reaches its maximum value which is

$$k_{n\,max} = \frac{A_{n\,max}}{A_1} = \frac{T}{4r_1(1 - r_1)\,T} = \frac{1}{4r_1(1 - r_1)} \tag{52}$$

It follows from equation (51) that at $r_1 = 0$, $k = n$ and with an increase in the number of spindles n, k increases proportionally according to a linear dependence (Fig. 133).

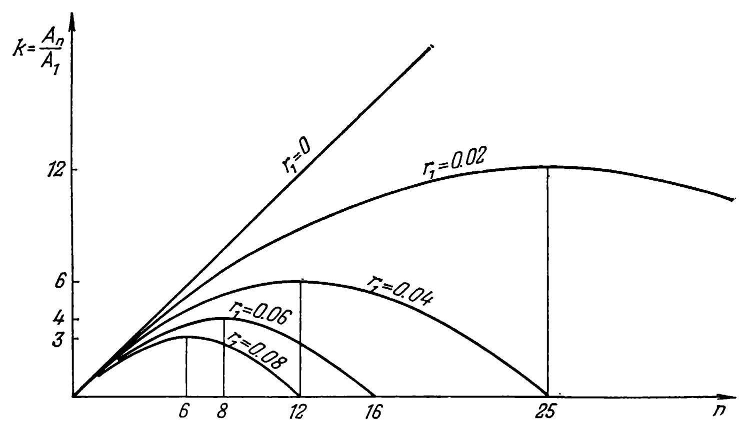


Fig. 133. Relative variation in the output of a machine tool of interlinked design with an increase in the number of spindles

From equation (45) it follows that $A_n = 0$ at

$$(1-r_1 n)\, nT = 0 \tag{53}$$

and since $n \neq 0$ and $T \neq 0$, then $A_n = 0$ when

$$1-r_1 n = 0 \tag{54}$$

and

$$n = \frac{1}{r_1} \tag{55}$$

It is evident that at $n = \frac{1}{r_1}$, the ratio $k = \frac{A_n}{A_1}$ is also equal to zero. The values A_n and k_n reach a maximum at a number of spindles $n = \frac{1}{2r_1}$ which is one half the number of spindles $n = \frac{1}{r_1}$ at which A_n and k_n are equal to zero.

Actual values of n and k for various values of r_1 are given in Fig. 133 and listed in Table 2.

TABLE 2

r_1	$n = \frac{1}{2r_1}$	$k_{n\,max} = \left(\frac{A_n}{A_1}\right)_{max}$	$A_n = 0$ and $k_n = 0$ at $n = \frac{1}{r_1}$	r_1	$n = \frac{1}{2r_1}$	$k_{n\,max} = \left(\frac{A_n}{A_1}\right)_{max}$	$A_n = 0$ and $k_n = 0$ at $n = \frac{1}{r_1}$
0.02	25	12	50	0.06	8	4	16
0.04	12	6	25	0.08	6	3	12

As machine tool engineering continues to develop, machine tools become more dependable in operation, and improved methods and facilities are devised for making readjustments in the setup. This leads to a reduction in the relative down time r_1 of the various spindles and at the corresponding positions or stations due to malfunctioning. As a result, the trend in design is to increase the number of spindles and stations.

CHAPTER 8

SETTING UP AUTOMATICS

Setting up an automatic lathe involves all the preparatory work required for the manufacture of a workpiece in accordance with the part drawing and specifications. In the sense here used, setting up includes planning the sequence of component operations; working out the calculation sheet for the setup; manufacturing the necessary tooling; setting up the kinematic trains to obtain the required speeds and feeds; and installing and adjusting the tooling.

8-1. Laying out the Sequence of Operations

The sequence of operations is worked out on the basis of the specific features of an automatic of the given type in respect to the cutting speeds, feeds and depth of cut; types of operations that can be performed at the various positions; facilities for setting up different speeds and feeds at the different positions; the possibilities of combining operations; the setup characteristics of the various working members; standard tooling at the various positions, etc. Data of this type are given in the service manual furnished with the automatic.

Along with these considerations, there are certain general rules for developing the tooling layout of general-purpose automatic lathes. These rules are based on the processing features and read as follows:

1. Overlap working operations wherever possible and make efforts to increase the number of tools operating simultaneously at each position.
2. Overlap handling operations with one another and with the working operations.
3. Begin finish machining only after roughing cuts are completed.
4. Do not permit substantial reduction of the rigidity of the workpiece before the roughing cuts have been completed.
5. Obtain accurate length dimensions along the workpiece by facing cuts from the cross slides, leaving an allowance of 0.5 to 1 mm for this purpose in turning from the end tool slide.
6. Speed up cutting-off operations which require much time, especially in solid stock, by widening the form tool so that it starts the cutoff, or by other preliminary grooving operations.

7. Break down form turning operations, wherever possible, into a rough and a finish cut.

8. Provide a dwell at the end of cross slide travel for cleaning up the surface to prevent out-of-roundness in form turning and to improve the surface finish.

9. In drilling a hole, first spot (centre) drill the work with a short drill of larger diameter.

10. In drilling a stepped hole first drill the largest diameter and then the next smaller diameters in succession. This reduces the total working travel required by all the drills.

11. To remove chips and facilitate drill cooling, drill deep holes in several cuts, withdrawing the drill each time. First drill to a depth $l_1 \leqslant 3d$; then to a depth $l_2 \leqslant 2d$ and finally to a depth of $l_3 \leqslant d$ (where d is the hole diameter).

12. If strict concentricity (or alignment) is required between external and internal surfaces, or stepped cylindrical surfaces, finish such surfaces in a single position.

13. To obtain equal machining times at all the positions of multiple-spindle automatics, divide the length to be turned into equal parts, or increase the rate of feed or the cutting speed at positions where a surface of longer length is to be turned.

14. Do not combine thread cutting with other operations. The calculated length of working travel should be increased by two or three pitches in comparison with the thread length specified in the part drawing, while the actual length of travel is reduced by 10 to 15 per cent of the calculated length by correspondingly reducing the radius of the cam at the end of the working travel movement. Thus, slide feed lags behind the movement of the tap or die along the thread being cut, excluding the possibility of stripping the thread due to incorrect feed of the slide by the cam. The tap or die should have a certain amount of axial freedom in its holder.

The tooling layout for all the operations consists of sketches drawn full size or to a convenient scale of the workpiece, cutting tools and holders in the relative positions they occupy at the end of the working travel movement. The grade and size of the tools, and the lengths of travel are indicated. These sketches are checked against the setting-up characteristics of the working members, and are also used to check whether the tools, holders and slides interfere with one another during operation. The sketches serve as well for selecting and designing cutting tools, holders and other tooling and to determine the radii and rises of the cam curves.

Speeds, feeds and depths of cut are assigned on the basis of the specifications of the given model of automatic as listed in the service manual. The cutting speed is limited, not only by the higher design tool life, but also by the insufficient rigidity of the spindle units which are usually crowded for

space, the combining of different cutting tools and operations in a single position, and the use of a single spindle speed in several positions when a lower cutting speed must be compensated for by a higher rate of feed.

The tooling layout serves as initial data for working out what is called the operation sheet for certain automatics and the cam design work sheet for others.

8-2. Determining the Time Required for Each Operation and Co-Ordinating the Working Travel Motions

In the operation sheet for a setup, not only the working travel, but also the idle travel movements of the working members as well as the motions of the auxiliary members are considered as separate operation elements.

To avoid impact of the cutting tools against the stock or blank at the end of the rapid approach, the length l of working travel is taken greater than the machining length l_1 by the amount Δ which is the length of approach at the rate of working feed. Thus

$$l = l_1 + \Delta \tag{56}$$

The value Δ is indicated in the service manual for the machine. In various types of automatics it ranges from 0.2 to 1 mm for end tool slides and from 0.1 to 0.5 mm for cross slides.

The *relative co-ordination of the working travel movements* is accomplished by finding the number of spindle revolutions n_i required for the given (ith) working operation element. At a constant speed of spindle rotation for all the working operation elements

$$n_i = \frac{l_i}{s_i} \text{ revolutions} \tag{57}$$

where l_i = length of working travel motion, mm, for the given operation element

s_i = rate of feed, mm per spindle revolution, in the given operation element.

If the spindle rotates at different speeds in the various operation elements, then

$$n_i = \frac{l_i}{s_i} C_i \tag{58}$$

where C_i is the equivalent revolutions factor. Thus

$$C_i = \frac{n_{spm}}{n_{spi}} \tag{59}$$

where n_{spm} = main speed of the spindle, rpm, in the setup
n_{spi} = spindle speed, rpm, in the given operation element.

The time required for the working operation element is

$$t_{wi} = \frac{n_i}{n_{spm}} \tag{60}$$

8-3. General Co-Ordination of the Cycle

The general cyclic co-ordination of the wo king and idle travel movements of the main working members and the motions of the auxiliary members is accomplished either in degrees or in hundredths of a revolution of the camshaft.

In carrying out these calculations, it is first necessary to determine Σk_{ii}, the number of degrees or hundredths of a camshaft revolution required for all (except the overlapped) idle and auxiliary (handling) movements. Next we find the number of hundredths or degrees for all the unoverlapped working travel movements:

$$\sum k_{wi} = 100 - \sum k_{ii} \tag{61}$$

or

$$\sum k_{wi} = 360 - \sum k_{ii} \tag{62}$$

where k_{wi} and k_{ii} are the number of hundredths (or degrees) for any given working and idle travel motion, respectively.

The total number of hundredths (or degrees) for all the working operation elements is distributed proportionally to the number of spindle revolutions n_i for the given operation element. Thus

$$k_{wi} = \frac{\sum k_{wi}}{\sum n_i} n_i \tag{63}$$

The number of hundredths or degrees k_{ii} for the idle travel of the main working members (approach and withdrawal of the tools), depending upon the length l_i of idle travel, is determined from tables or cam templates for idle travel curves as $\beta_i = f(l_i)$. These data are given in service manuals for certain wide ranges of cycle times T_c and output values. It is assumed for a tentative determination of the cycle time that all the idle travel movements and auxiliary motions constitute a certain percentage of the total time T_w of all unoverlapped working travel movements. According to equation (60), the time T_w is determined as

$$T_w = \sum t_{wi} = \frac{\sum n_i}{n_{spm}} \tag{64}$$

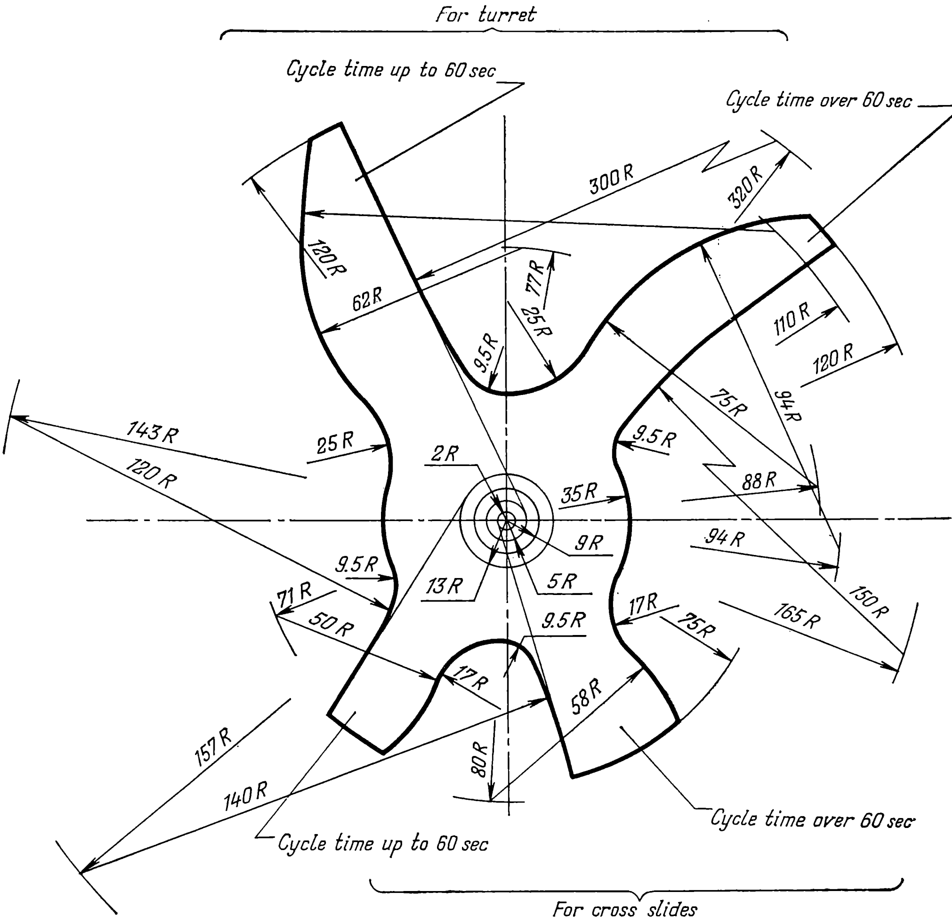


Fig. 134. Cam template for drawing the curves for idle travel movements of the slides of the model 1A136 automatic screw machine

For example, in a turret-type automatic screw machine, the tentative cycle time T_{ct} is taken as

$$T_{ct} = (1.25 \text{ to } 1.30)\, T_w \tag{65}$$

The number of hundredths or degrees k_{ii} required for the auxiliary motions of the working members is either given in the service manual of the automatic directly in hundredths or degrees, or it is calculated from the time t_{ii} indi-

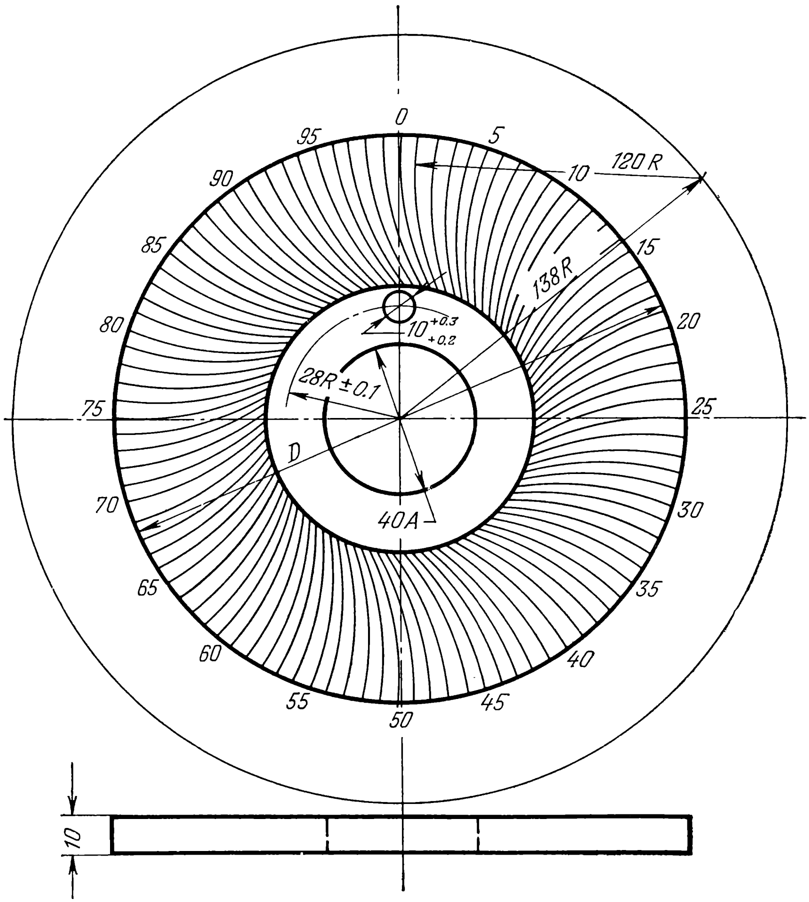


Fig. 135. Cam blank for the turret slide

cated in the service manual for auxiliary motions, using the relationship

$$k_{ii} = \frac{t_{ii}}{T_{ct}} 100 \qquad (66)$$

where T_{ct} is the tentative cycle time as determined from a formula similar to (65).

Turret indexing through $\frac{1}{6}$ revolution requires a definite time in automatic screw machines (for example $\frac{2}{3}$ sec in model 1A136 and $\frac{1}{2}$ sec in model

1A112). The end of turret indexing coincides in time with the end of turret slide withdrawal along the drop curve of the lead cam (see p. 65). Consequently, the time for withdrawal is not taken into account, being referred to turret indexing. This is taken into consideration, in part, by the table in the service manual which lists the number of hundredths required for turret indexing in accordance with the radius on which indexing is completed. The number of hundredths is increased with a decrease in the radius. This increase in the number of hundredths is also called for by the conditions for drawing the cam curve when the roll diameter is taken into account.

The number of hundredths required for turret indexing can be accurately determined by superimposing the cam template (Fig. 134) on the blank for the turret slide cam (Fig. 135). For this purpose, concentric circles are drawn on the cam blank with radii from 40 to 115 mm in 5-mm intervals.

After calculating the sum of hundredths Σk_{ii} for all the unoverlapped idle travel and auxiliary motions according to formulas (61), (62) and (63), the number of hundredths (or degrees) k_{wi} of camshaft revolution is calculated for each working travel movement.

8-4. Determining the Cycle Time

In automatics of the first and third structural groups (see p. 43), the total number of spindle revolutions required to machine a workpiece n_{wp} is determined from the formula

$$n_{wp} = \frac{100 \sum n_i}{\sum k_{wi}} \tag{67}$$

Then the time required to make one workpiece, or cycle time, is

$$T_c = \frac{60 n_{wp}}{n_{spm}} \text{ sec} \tag{68}$$

The value obtained here is then rounded off to the nearest value available in the table of change gears given in the service manual for setting up the speed of camshaft rotation. If the value of T_c obtained here differs so greatly from the tentative value taken from formula (65) that this substantially affects the values of k_{ii}, taken previously from the table or calculated by formula (66), it may be necessary to make recalculations.

The calculated values of camshaft rotation k_{wi} and k_{ii} are written into the operation sheet, and a cyclogram, or timing sheet, of the rectangular or circular type is constructed (see Fig. 37).

Unlike circular cyclograms, the rectangular type can be used to determine the relative movements of the working members, and to check for interference if these movements are plotted in a definite scale.

The general method described above for calculating the number of hundredths (or degrees) required for the working travel movements (k_{wi}) and the idle and auxiliary motions (k_{ii}) is based on the design of automatic lathes of the third group (turret-type automatic screw machines). These calculations can be simplified for automatics of the first and second groups.

8-5. Working out the Operation Sheet for Multiple-Spindle Automatics

In these machines the machining time is the same at each position. To bring this time to a minimum, the operation element requiring the most time is divided between two or three positions. The spindle speed n_{sp} rpm is established for the operation of turning with a form tool.

Multiple-spindle automatics belong to the second structural group in which idle travel and auxiliary (handling) motions are accomplished at rapid rotation of the camshaft through a constant angle β (in model 1240-6, $\beta = 220°$, see Fig. 37) during the constant time t_i allotted for idle travel of the slides and for auxiliary motions (in model 1240-6, the time $t_i = 2.1$ sec).

First we determine the number of spindle revolutions n_i during the time t'_w for the longest operation. Thus

$$n_i = \frac{l}{s} \tag{69}$$

where l = actual length of travel, mm, at the rate of working feed (see below)
s = actual rate of feed, mm per spindle revolution.

The time required by the longest operation is

$$t'_w = 60 \frac{n_i}{n_{sp}} \text{ sec} \tag{70}$$

To this we add the time t_{cl} required for the engaging of the camshaft high-speed clutch which takes place during camshaft rotation at working travel speed through the angle β_{cl} (in model 1240-6, angle $\beta_{cl} = 4.5°$).

Working travel rotation of the camshaft takes place in angle $\alpha_w = 360° - \beta$, while the longest operation corresponds to angle $\alpha_w - \beta_{cl} = 360° - (\beta + \beta_{cl})$.

The time required for the working travel rotation of the camshaft is

$$t_w = t'_w \frac{\alpha_w}{\alpha_w - \beta_{cl}} = 60 \frac{n_i}{n_{sp}} \frac{360° - \beta}{360° - (\beta + \beta_{cl})} \text{ sec} \tag{71}$$

The calculated lengths of travel of the tools, $l = l_1 + \Delta$, are obtained in automatics employing permanent sets of cams by adjusting the lever system of transmission from the cam to the slide. For the selected cam with the required rise, the actual lengths of working travel and the actual feeds

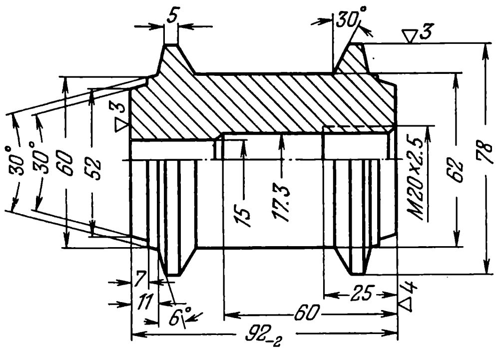


Fig. 136. Roller

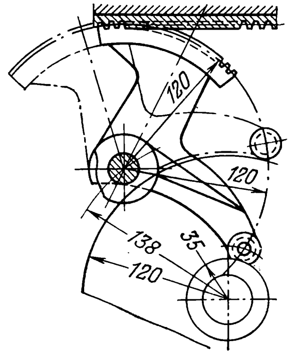


Fig. 137. Lever of the turret slide plate cam

in mm per revolution are calculated according to the data of the service manual, taking into account the transmission ratios of the lever systems.

The number of spindle revolutions required for the unoverlapped working operation elements is calculated for each position by using formula (69). The value n_i should not vary greatly for the different positions. The change gears of the camshaft drive are selected according to the value of n_i.

The tooling layout and operation sheet for machining the roller shown in Fig. 136 in the model 1265 automatic is given in Table 3.

TABLE 3

Tooling Layout and Operation Sheet

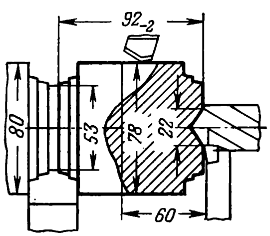
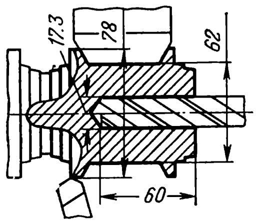
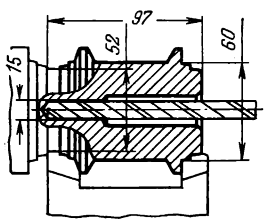
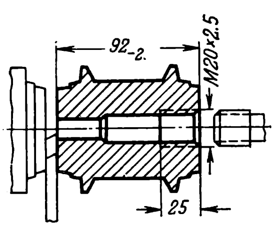
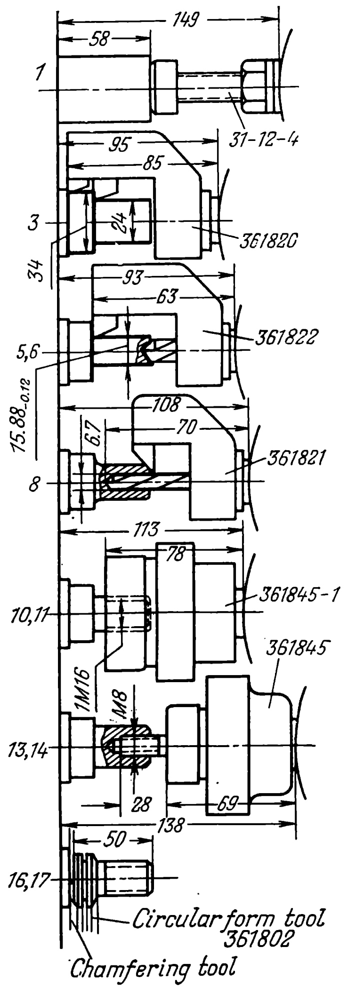
Operation sequence	Operation elements	Calculated machined length, mm	Actual working travel, mm	Feed, mm per rev	Spindle revolutions corresponding to working travel	Accepted number of revolutions
	Spot drill to dia 22 and turn dia 78	50.5	65	0.17	383	394
	Face end and turn shoulder	20	22.8	0.06	380	394

Table 3 (continued)

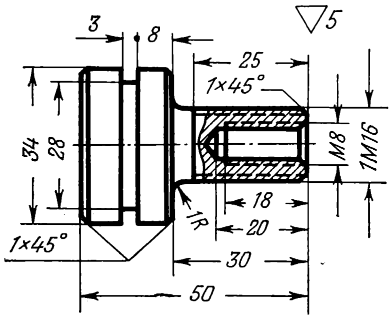
Operation sequence	Operation elements	Calculated machined length, mm	Actual working travel, mm	Feed, mm per rev	Spindle revolutions corresponding to working travel	Accepted number of revolutions
	Turn dia 78 and drill hole dia 17.3	62	65	0.17	383	394
	Form-turn groove to dia 62	11	13.7	0.036	380	394
	Drill hole dia 15	38		0.1	181	
	Form-turn hubs	10.5	15.5	0.041	378	394
	Tap thread M20 × 2.5	30		2	23	
	Cut off	20	25.6	0.068	376	394

TAB

Tooling Layout and Operation

Tooling layout	Tools			Slide	Order of operation elements	
	cutting	auxiliary	measuring			
	Straight turning tool P 17-4 3 pcs Chamfering tool P 17-2 Circular form tool for groove and chamfer Chamfering tool of vertical slide P 17-26 Cut-off tool of rear slide P 17-18	Stock stop 31-12-4 Holder 361820 Holder 361822 dia 25 Bushing 361834 dia 10 Holder 361821 dia 25 Bushing 361833 dia 6.7 Die-holder 361845-A Tap-holder 361842 $d=6$ Rear slide holder 361811, 361813	Steel rule Vernier caliper Ring thread gauge 1M16 Plug thread gauge M8	Tur-ret	1. Feed stock to stop 2. Index tur-ret 3. Turn dia 34 and 24 4. Index tur-ret 5. Turn dia $15.88_{-0.12}$ and spot drill 6. Dwell 7. Index tur-ret 8. Drill 6.7; chamfer 1×45° 9. Index tur-ret	

Sheet for an Automatic Screw Machine



Travel (cam throw), mm	Feed, mm per rev	Spindle revolutions: working travel	Spindle revolutions: for calculations	Plate cam, Hundredths: working travel	Plate cam, Hundredths: idle motions	Plate cam, Hundredths: start	Plate cam, Hundredths: finish	Plate cam, Radii: start	Plate cam, Radii: finish
					1	0	1	64	64
					2.5	1	3.5	63	63
54	0.12	450	450	15.5		3.5	19	64	118
					3	19	22	88	88
31	0.15	207	207	1.5	—	22	29.5	89	120
					0.5	29.5	30	120	120
					4	30	34	81	81
23	0.066	350	230	8		34	42	82	105
					4	42	46	71	71

Machine		
Type	Model	Inv. No
Automatic screw machine	1A136	
Coolant	Sulphurized oil	
Material	Grade	Hardness
Steel	A12	—

Bar stock
dia 36 × 3,000

Change gears			
A	B	C	D
20	80	60	45

Setting distance — turret to spindle nose	173

Spindle reversal cam positions			
Backward		Forward	
I	2.5		
	19		
III	30		
		IV	42
		V	60
VI	97		

Stock feed	0

TABLE 4

Tooling layout	Tools			Slide	Order of operation elements	
	cutting	auxiliary	measuring			
	L.H. drills 6.7 and 10 Round threading die 1M16 Tap M8	Front slide holder 361802 Cams: turret slide T front slide F rear slide R vertical slide **V**		Turret	10. Cut thread 1M16×1.5	
					11. Back off die	
					12. Index turret	
					13. Tap thread M8×1.25	
					14. Back out tap	
					15. Index turret	
				Cross	16a. Turn groove to dia 28	
					Front slide	chamfer 1.5×45°
						dwell
					Rear slide	cutoff
						withdraw cut-off tool
					16b. Vert. slide	chamfer 1.5×45°
						dwell
					Total	

(continued)

Travel (cam throw), mm	Feed, mm per rev	Spindle revolutions: working travel	Spindle revolutions: for calculations	Plate cam, Hundredths: working travel	Plate cam, Hundredths: idle motions	Plate cam, Hundredths: start	Plate cam, Hundredths: finish	Plate cam, Radii: start	Plate cam, Radii: finish
28	1.5	19	200	7	—	46	53	72	97
28	1.5	19	200	7	—	53	60	97	72
					4.5	60	64.5	50	50
21	1.25	17	60	2	—	64.5	66.5	54	73
21	1.25	17	60	2	—	66.5	68.5	73	54
					(2)	(97)	(99)	50	50
3.5	0.03	117	(180)	(6.5)		(68.5)	(75)	57.5	61
					(0.5)	(75)	(75.5)	61	61
18.3	0.035	525	810	28.5	—	68	97	56.7	75
					3	97	100	75	35
2.5	0.03	84	(129)	(4.5)	—	68.5	(73)	57.5	60
					(0.5)	(73)	(73.5)	60	60
			2,217	77.5	22.5				

Operation element	3 and 5	8	10	12	16
Equivalent revolutions factor	1	0.665	10.5	3.5	1.54
Spindle speed, rpm	1,050	1,580	100	300	680

Speeds and feeds			
Machining	Turning, cut-off	Drilling	Cutting thread
Cutting speed	118.5; 75	33.3	5; 7.5
Spindle speed, rpm	1,050; 680	1,580	100; 300
Main speed of the spindle (for calculations), rpm			1,050
Total spindle revolutions for working travel			2,217
Total spindle revolutions for idle motions			643
Total spindle revolutions for making one workpiece			2,860

Figuring Cam Radii

In respect to its configuration, the lead cam of the turret slide in an automatic screw machine is one of the most complex cams. The transmission ratio from the cam to the slide is equal to unity (Fig. 137) so that in cam rotation the displacement of the turret slide is equal to the change in the cam radius. For each position of the slide

$$L_j + R_j = \text{const} \tag{72}$$

where L_j = distance from the end face of the collet to the edge of the turret
R_j = corresponding radius of the lead cam.

The tooling layout (see Table 4) is used to determine the dimension L^{II} for the position of the turret at the end of each working travel movement when the roll is at the corresponding cam radius R^{II}. For the operation element with the minimum value L^{II}_{min}, the cam radius R^{II}_{max} is taken equal to 120 mm or one half of the diameter of the cam blank. This is done because in operation with large cam radii the load factor is reduced and there is less danger of jamming.

The minimum value in the setup of Table 4 is $L^{II}_{min} = L^{II}_{5,6} = 93$ mm for the 5th and 6th operation elements. Thus, $L^{II}_{5,6} + R^{II}_{5,6} = 93 + 120 =$ $=$ const $= 213$ mm.

Therefore, in the given setup, $L_j + R_j = 213$ mm for all the positions of the turret, and the cam radius at the end of each working operation element is

$$R^{II}_j = 213 - L^{II}_j \tag{73}$$

with the exception of the thread cutting (tapping) operation in which case the calculated radius at the end of working travel R^{II} is reduced by 10 per cent of the travel length to enable the tool (tap) to be fed by the thread being cut, leading the slide to avoid stripping of the thread due to incorrect slide feed.

The bar stock is fed out to the stop at the cam radius $R^{II}_1 = 64$ mm, equal to the radius R^{I}_3 at the beginning of the next working travel movement, so as to withdraw the turret less after indexing. Using formula (72), we find $L^{II}_1 = 213 - R^{II}_1 = 213 - 64 = 149$ mm and correspondingly set up the stock stop.

According to the service manual of model 1A136, the maximum travel (and turning length) of the turret is $h = 80$ mm. The maximum distance from the end face of the collet to the edge of the turret (specified turret dimension) $Q = 188$ mm and the minimum distance is $P = 64$ mm. For a specified turret dimension $Q = 188$ mm, $L^{II}_{min} = Q - h = 188 - 80 = 108$ mm when $R^{II}_{max} = 120$ mm, and $L^{II}_{min} + R^{II}_{max} = \text{const} = 108 + 120 = 228$ mm.

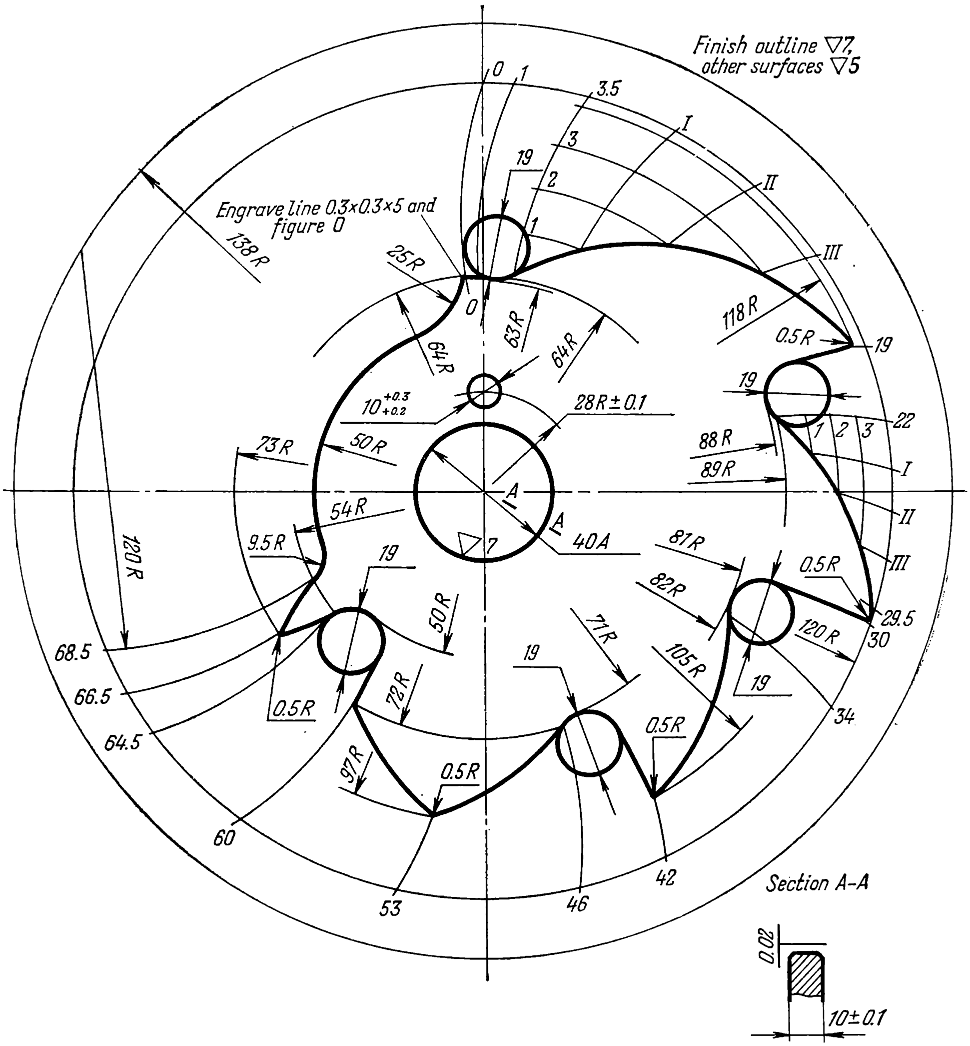


Fig. 138. Turret slide cam. The curves for the working travel and for backing the tap out and the die off are Archimedean spirals; the withdrawal curves (roll drop) are constructed to a template. The tolerance is 0.1 mm on the working travel curves

The maximum value of the constant in the equation is limited: $L_j + R_j = $ $= \text{const} = 228$ mm. Hence, if $L^{II}_{min} > 108$ mm, then correspondingly R^{II}_{max} is taken less than 120 mm.

If $L^{II}_{min} < 108$ and $Q = L^{II}_{min} + h + 80 < 188$ mm, then R^{II}_{max} is taken equal to 120 mm and the turret slide is set up by adjustment (in reference to the rack). The holders are set up in the tool holes of the turret. This is done when $L^{II}_{min} \geqslant 64$ mm.

The design of the cam for the operation sheet in Table 4 is shown in Fig. 138.

8-6. Group Setups for Automatic Lathes

The following approximate time requirements for setting-up operations have been established on the basis of automatic screw machine practice:

1. Adjustments in the settings require 0.5 to 1.5 hours per shift.
2. Changing over the machine for another part similar in construction and processing method, without cam changes, requires 0.5 to 1.3 hours if the collet and pusher must be replaced and 0.2 to 0.5 hour if the same collet and pusher can be used.
3. A complete change-over of the automatic in which cams, collet, stock pusher and tools are changed requires from 4 to 8 hours.

The maximum amount of time is lost in a complete change-over. If the workpieces are segregated into groups, or "families", so that in changing over from one workpiece to another within the same group no cam changes are required, then change-overs within groups can be done in one-tenth to one-twentieth of the time needed for a complete change-over. This enables automatics to be employed efficiently in small-lot production.

Segregation into groups is accomplished by classifying all the workpieces that are to be machined according to the type and size of the automatic lathe in which they can be most expediently machined: automatic cutting-off machine, Swiss-type automatic screw machine, turret-type automatic screw machine or multiple-spindle automatic bar machine. Then the workpieces in each class are subdivided into groups so that all the workpieces in a group can be machined with a single set of cams (Fig. 139).

A general sequence of operations is developed for each group. On the basis of this sequence, a tooling layout is worked out for each workpiece which, in turn, is used to determine the lengths of the working travel movements. Next, the overall dimensions of the integrated group workpiece are established from the maximum lengths of the working travel movements.

The dimensions are denoted by letters on the sketch of the integrated workpiece in the tooling layout, and the maximum and minimum values are listed in a table as shown in Fig. 140.

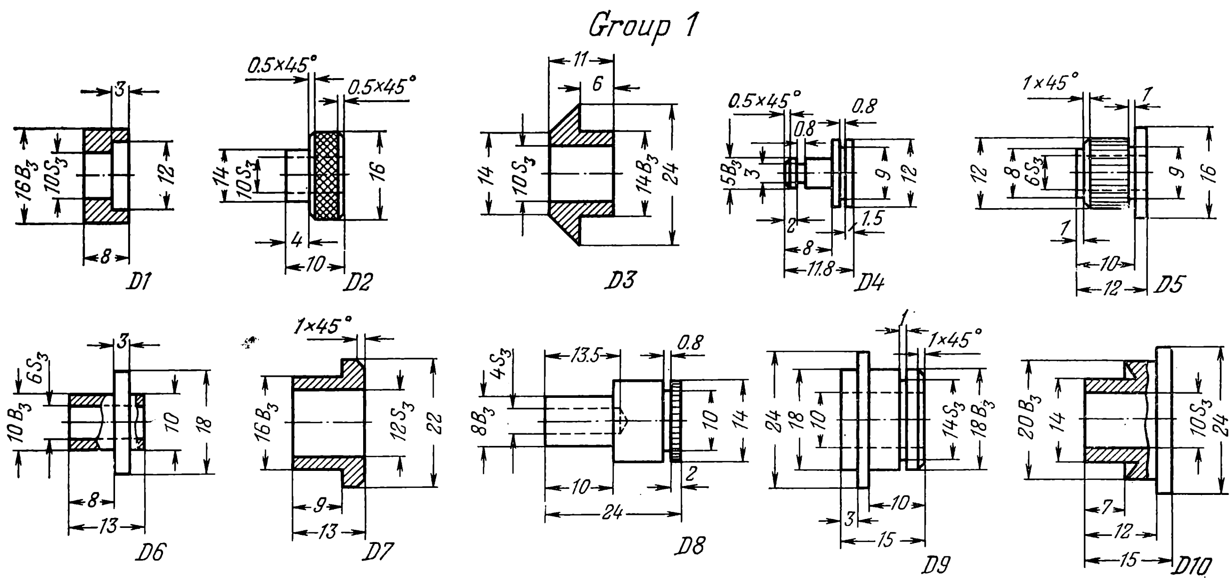
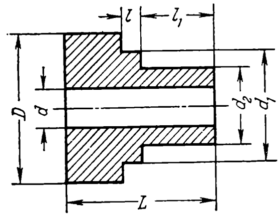


Fig. 139. Group of workpieces machined in the model 1118 automatic



D	d	d_1	d_2	$\frac{D-d}{2}$	$\frac{D-d_1}{2}$	L	l	l_1
9-26	5-15	7-20	5.8-17	3.5-10	0-7	9-15.5	2-3.5	4.5-10

Fig. 140. Integrated workpiece

All calculations for the setup are based on the integrated workpiece, the following working operation elements being provided in the sequence (for the workpiece shown in Fig. 140): (1) feed stock to stop; (2) spot drill; (3) drill dia d and rough turn dia d_1; (4) turn dia d_2; (5) bore dia d and turn dia d_1 from the rear slide; (6) face from the rear slide and cut off from the vertical slide.

The following criterion enables the economic feasibility of a group setup for an automatic to be tentatively assessed:

$$\frac{A}{n}+B \leqslant \left(\frac{a+b}{n}+T\right) \text{ min} \tag{74}$$

where A = time, min, required for setting adjustments on the automatic in fulfilling the annual task if a group setup is used
n = number of workpieces machined to fulfill the annual task
B = time, min, required to make the integrated workpiece
a = time, min, required to make the cams for the annual task if the workpieces are to be machined with individual setups
b = time, min, required for complete change-overs for the annual task if the workpieces are to be machined with individual setups
T = time required to make the workpieces with individual setups, min.

In a group setup, the machine is changed over without changing the cams by making adjustments in the tooling. The processing capacity of such tooling has been extended in up-to-date automatic screw machines by the application of swing stops which free one tool hole in the turret, and the use of two or three vertical slides, arranged fan-like, instead on a single vertical slide.

This last measure increases the number of cross slides to five, of which the front slide may be of the compound type, providing for both longitudinal and cross feeds.

If the same set of cams is used for a group of workpieces, it becomes necessary to effect slide approach for workpieces with small lengths of working travel along the working feed curves of the cams. This leads to the necessity of resorting to high-speed rotation of the camshaft and of regulating the travel of the slides by setting up the transmitting members between the cams and the slides.

Special attachments with change gears are used to cut threads with a single-point threading tool or with a chaser. The cross slide cams are made of disks of the same diameter so that any slide can be used to actuate any cross slide. The cross slide cams for a group setup are installed on the machine in the form of a single general block.

The design measures mentioned above for extending the possibilities of group setups have been applied, for example, in the model A50A automatic screw machine (made in Czechoslovakia) and the Manurhin automatics, model TR32-60B (made in France).

MACHINE TOOLS WITH NUMERICAL CONTROLS

CHAPTER 9

BASIC CONCEPTIONS, DEFINITIONS AND PRINCIPLES

The most successful development of the mechanization and automation of production processes in the engineering industries can be found in large-lot and mass production plants where wider and wider use is being made of general-purpose and special automatic and semiautomatic machine tools, automated unit-built machine tools, automatic transfer machines, automated production lines and departments, and even completely automated plants.

Small-lot and piece production, in which about half the available machine tools are engaged, and large-lot and even mass production, in which the product undergoes frequent and complete changes (as in the aircraft and other special branches of industry), are often difficult to automate by ordinary means. The manufacture of special tooling and the time-consuming setting up of the machine tools, or the design of special automatic machines is often unprofitable in such cases.

Recently developed machine tools with automatic control, such as tracer-controlled machines and automatic and semiautomatic machine tools with a programmed operating cycle and in-travel control of the operative units, have been intended for use in small-lot production.

Nevertheless, it still takes considerable time to change over these machines for handling parts of different configuration. Moreover, it is necessary to manufacture cams, templates or blocks of stops and to store them in large amounts. All these factors essentially reduce the efficiency of such equipment for small-lot production. As far as piece production is concerned, even of the repeating type, here such machine tools are entirely unsuitable.

The main type of equipment used in small-lot and piece production is still the general-purpose machine tool in which the operator is included in the control system; his mental and physical labour participates in the shaping of the workpiece. It should be noted that in some cases an increase in labour productivity in lot production leads to considerable fatigue of the operator due to the monotony of the manual operations, especially if the same operation is performed on large lots. The automation of small-lot and piece production requires the development of general-purpose templateless automatic

machine tools with convenient, flexible facilities for feeding the needed workpiece machining information into the machine (preparing the programme) and with means for easily storing this programme for any length of time to make possible its repeated use. At the present time, this purpose is best served by *numerically controlled machine tools.*

The novelty of the questions being considered and the lack of an established terminology give rise to the necessity for formulating the basic principles and presenting the fundamentals of the programme and numerical control systems of present-day machine tools.

9-1. General Principles

Almost any automatic machine tool can be classed as one with programme controls. Indeed, in cam- or tracer-controlled machines, the programme of operation of the operative members (units) is predetermined by the profile of the cams or templates. The programme of operation can also be given by a master switch which controls the operating cycle of the machine tool, the travel of the units being restricted, for example, by trip dogs operating limit switches. Nevertheless, we continue to call this kind of machine an automatic or semiautomatic machine tool as before. In these machines, the information-carrying, or storage medium (cam, template, system of definitely positioned trip dogs and limit switches, etc.) is the programme for shaping the workpiece, physically realized in a definite scale. This medium is kinematically linked to the operative units (in a definite manner for the given setup) either directly (camshaft-type automatics) or through a system of amplification and control (tracer-controlled machine tools). Such control is insufficiently flexible, since it is necessary to change the parameters of the kinematic train in order to vary the length of travel of the operative unit (by changing cams or templates, changing transmission ratios, adjusting trip dogs and stops, etc.).

In considering numerically controlled machine tools we shall deal only with the questions of control that are associated with the motions of the operative units for shaping the workpiece, or with positioning motions of the workpiece and tool, since only these questions are specific to machine tools of this type.

Cycle control, changing of the speeds and feeds, and tool changing are automated by conventional methods. Even in cases when these functions are accomplished by means of punched cards, punched tape or selector switches, the structure of the controls remains the same as in conventional automatic machine tools with master switches, though these machine tools can rightly be called *machine tools with a programmed control of the cycle, speeds and feed or tool changing.*

9-2. Concept of Numerical Controls for Machine Tools

Numerical control is based on the use of numerical data for directly controlling the positions of the operative units of a machine tool in the course of a machining operation. In the following we shall deal with numerical control of operative units travelling with rectilinear motion, though this type of control is equally suitable for operation with any other kind of motion.

In a system of numerical control, the operative units of the machine tool are controlled in their motion by numbers which determine the shape and size of the workpiece and which are consecutively fed into the control system. In the intervals between command signals, the shape of the workpiece is determined only by the ways of the travelling operative units and their relative speeds of motion. In positioning motions, it is not necessary to specify the path of motion of the operative unit; only its final position is controlled.

For the sake of brevity, machine tools with numerical controls will be called N/C machine tools in the following. These machines are characterized by a system of controls enabling them to be quickly changed over for another job without changing or rearranging any mechanical elements, i.e., by templateless remote controls. The magnitudes of displacement of the operative units, determining the shape and size of the workpiece, are fed into the control system in the form of numbers which represent the shape of the path, and the magnitude, direction and velocity of this motion. Such control is more flexible than tracer control; it suffices to vary the information fed into the control system of the machine tool and the geometric parameters of the workpiece will be changed accordingly.

As a rule, the requirements made to a part being manufactured are specified by the part drawing. For an N/C metal-cutting machine tool, information in the form of a drawing must be transformed. It must be given a form that will allow the machine tool to be automatically controlled.

The programme of an N/C machine tool is usually prepared beforehand. On the basis of the part drawing and the operation sheet, tables are compiled listing the magnitudes, directions and rates of feed of the consecutive motions of the operative units. These data are registered, for example by setting selector switches on the control panel, or they are recorded by punching holes in the storage medium (tape or cards) or are remembered by some other technique. The input medium is fed into a special reader on the machine tool. Numerical information, fed by this procedure into the machine tool, is transformed into the corresponding movements of the operative units. Automatic preparation of programmes for N/C machine tools by means of electronic computers is common practice at the present time.

Hence, we shall call *numerically controlled machine tools ones in which numerical or symbolic information, representing the magnitudes, speeds and*

directions of motion of the operative units participating in the machining of the workpiece or in positioning the workpiece and/or cutting tool is fed into the system of automatic control. Discrete programming is characteristic of N/C machine tools, since the programme is determined by numerical information.

The design of N/C machine tools, their manufacture and especially their operation require a different approach than to conventional machine tools. Thus, for example, less skill and mastery of his trade is required of an operator tending an N/C machine tool in small-lot or piece production than of one tending general-purpose equipment. On an N/C machine the operator is converted into an operator-observer whose functions are more of an auxiliary nature. He prepares the tooling and the machine for operation, sets up the blank and removes the finished workpiece (if loading and unloading have not been automated), and feeds the data medium into the control unit. Along with a reduction in the skill required of an operator, in the sense this term is usually understood, his knowledge of equipment of the new type, of the machining process, mechanics, and especially electrical engineering and electronics should be much higher, approaching that of a technician. N/C machine tools are not so complex mechanically as they are full of electrical equipment and have an extensively branched control system. The operator should learn to eliminate by himself minor faults or disorders in the operation of various units which, unfortunately, cannot be completely excluded as yet.

It should be pointed out that N/C machine tools have not been very widely used in industry until only recently. The main aim of research in N/C machine tool engineering at the present time is to increase the dependability of these machines. This is the most characteristic feature of N/C machine tool design and manufacture.

A number of factors led to the development of N/C machine tools. One was indicated above and constitutes the continuous development of the engineering industries, in which a further growth in production capacity may be impeded in many cases by the unadaptability of conventional means of automation to small-lot and piece production, or even to large-lot and mass production if the product is frequently changed.

The development of N/C machine tools, using only previously known methods of automation, for instance, relay circuits, could not offer satisfactory results due to the unwieldiness of such circuits and their low dependability. The design of such machine tools became possible only as a result of the achievements in the fields of electronics, semiconductor and computer engineering.

The development of N/C machine tools was also stimulated in a considerable degree by theoretical investigations in engineering cybernetics and information theory. In cybernetic systems, *information* may be, in particular, a communication indicating a change in the characteristics of the controlled

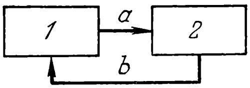


Fig. 141. Two-way interlink in an automatic control system: *a*—direct control signals; *b*—feedback signals; *1*—controlling unit; *2*—controlled unit

unit—machine tool or other machine—and in the external disturbances acting on the unit. Extensive theoretical investigations in this field have been conducted in the Soviet Union and other countries.

9-3. Conception of Feedback

As a rule, there is a two-way interlink between the controlling and controlled units in the form of signals of direct control action and feedback (Fig. 141). *Direct control signals* indicate how the controlled unit is to act, while feedback indicates how it actually acts and what is to be done to make its behaviour correspond to the given (required) behaviour.

A *feedback signal* is received on the input of an arrangement in which the difference between the given and actual states (for instance, travel values)—the error—is used as a new input signal which makes the controlled unit change (move) so that the actual state (movement) of the arrangement comes nearer and nearer to the given value. In other words, feedback transmits a signal which corrects the process. If the feedback signal *is summed up with* the control (direct) signal it is said to be *positive*; if it *is subtracted from* this signal it is called *negative feedback*.

In prediction systems with feedback, an *extrapolating* or *predicting compensator* is incorporated in the control system. A system can have "informative" feedback. An example of such feedback is the checking of the condition of an icy highway by the driver of an automobile when he receives his information by testing whether the car skids or not.

The reception, transformation and transmission of information takes place in all automatically controlled systems.

Information can be accumulated or stored. This principle is the basis for the whole process of training persons or developing machines which facilitate (mechanize) mental labour.

The same information may be represented in various ways. For the simplest example, the number five can be represented by either Arabic or Roman numerals (5 or V), vertical lines (|||||), words in different languages, etc. The vital features in this representation are its unambiguity and distinguishability. Thus, to denote figures from 0 through 9, it is necessary to have ten different designations. It does not matter what symbols are used to express the figures; the only important requirement is that they should not coincide.

The representation of various communications in distinguishable ways (in the form of definite signals) is called the *coding* of the information, and the method of representing the information in the distinguishable form is called the *code*.

9-4. General Structures of N/C Machine Tools

Almost any control circuit for an N/C machine tool can be represented by the following block diagram (Fig. 142).

Unit No. 1 is for compiling and recording the programme. The source of information in compiling the programme is usually a drawing of the part to be manufactured. From the dimensions or tabular data given in the drawing, the programmist, usually a processing engineer, works out a table in which either the absolute values of the co-ordinates of the programmed points on the machined sections or the increments of the co-ordinates of these points are indicated. The direction of motion of the operative unit and its speed are also set down. Next, by means of a perforator, this, in most cases, numerical information is recorded in the coded form on the storage medium, and we obtain a punched card or tape. In some cases there is no input medium and the information is fed directly into the machine tool by means of multiple-position selector switches (usually 10-position switches fixing any figures from 0 through 9) located on the control desk. As mentioned above, in many cases the programme is prepared by a whole set of automatic equipment including an electronic computer.

Unit No. 2 verifies the programme. It is advisable in any case to check the compiled programme. This can be done by visual inspection of the storage medium or by checking the positions of the switches on the control desk. Sometimes, an idle travel run around the sections to be machined is resorted to, or a test piece is machined. In other cases, special automatic devices may be employed to check the storage medium. There is no difficulty in checking comparatively simple programmes by recording them twice and comparing the two records.

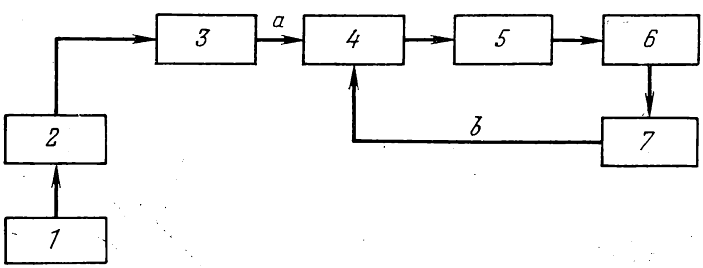


Fig. 142. Typical block diagram of the numerical controls of a machine tool:
a—control signals; *b*—feedback signals; *1*—compiling and recording the programme; *2*—verifying the programme; *3*—feeding the programme into the machine tool; *4*—converting the programme, controlling the machine and checking response; *5*—operative unit drive; *6*—operative unit; *7*—measuring element or feedback transducer

Unit No. 3 feeds in the programme. The programme may be infed by setting switches or by means of a mechanism for reading the programme if it has been recorded on some storage medium.

Unit No. 4 is for converting the programme, for controlling the machine and for checking whether the required movements have been made. This unit controls the drive of the operative unit in accordance with the information received from the storage medium and the operative unit.

Unit No. 5 is the drive of the operative unit. Such drives commonly are power servosystems with a considerable power amplification factor.

Unit No. 6 is the operative unit. This may be the carriage of a lathe or the table of a drill press or milling machine.

Unit No. 7, the feedback transducer, checks the position of the operative unit. Its signal corrects the position of the unit. The presence of a sensor or transducer makes the system a closed-loop one. Sometimes, open-loop control systems, having no transducer, are used.

9-5. Classification of, N/C Systems According to Their Processing Features

In accordance with their processing purpose, N/C systems are divided into *finite positioning control systems* and *continuous or contouring control systems*.

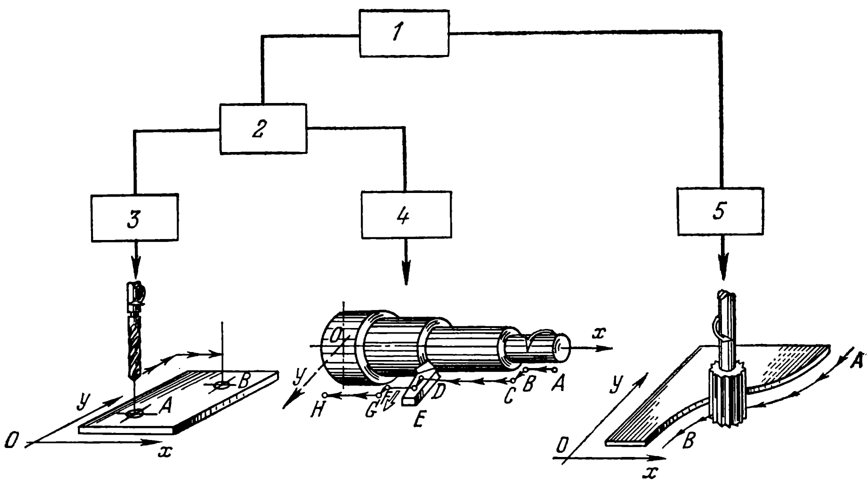


Fig. 143. Classification of N/C systems according to their processing features:
1—N/C; *2*—finite positioning controls; *3*—point-to-point systems; *4*—straight-line systems; *5* — contouring systems

In turn, finite positioning control systems can be classified as:

(a) Systems for setting co-ordinate dimensions without definite linear motion between the points machined (the operative units can travel simultaneously but with no interrelation). In machine tools with such N/C systems the processing operation is performed after the operative unit has reached its co-ordinate position. These may be called *point-to-point systems.*

(b) Systems with rectilinear consecutive movement of the operative units, parallel to the directions of travel of the operative units, from point to point in the process of machining the workpiece. These can be called *straight-line systems.*

In the continuous control systems, there are continuous, simultaneous and co-ordinated, i.e., interrelated, motions of the tool and workpiece along different co-ordinate axes, thereby enabling curvilinear contours or surfaces to be machined. It is used where the position of several slides must be controlled so that their relative positions and velocities must be established at every point and continuously throughout the operation.

Continuous control systems can be called contouring systems.

Representative machine tools using point-to-point N/C systems are drilling and boring machines; in lathes and vertical boring mills, as a rule, straight-line systems are used, while milling machines are usually equipped with contouring N/C systems (see Fig. 143).

9-6. Field of Application of N/C Machine Tools

In addition to the above-mentioned applications—in small-lot and piece but repeating production and in lot production with a frequently changed product—a number of fields can be named in which N/C machine tools can find efficient application. Such are the manufacture of parts of complex configuration (for example, the blades of reaction turbines, propellers, etc.) where kinematically complex motions are required; the manufacture of parts in experimental or pilot production requiring complicated tooling which would be unwanted for making the parts in general-purpose machines; and cases when it is necessary to arrange the workplace remote from the machine tool in production which is a health hazard.

In addition to the main advantageous feature, the flexibility of such systems, it should be noted that the level of production management is substantially raised in a plant using N/C machine tools because production engineering and tooling have been automated to a great degree. This is especially important in small-lot and piece production.

Along with their obvious advantages, N/C machine tools have one essential drawback as yet—their exceptionally high cost. For example, the cost of a drill press is increased approximately fivefold if it is equipped with an N/C system. At the same time, according to certain data, the expenditures in

purchasing expensive N/C milling machines are justified after two years of operation. In calculating the economic feasibility of applying N/C machine tools, it is necessary to take into consideration all the production engineering activities required to manufacture the parts, as well as the time necessary for changing to the production of other types of parts with the existing equipment and when N/C machine tools are available.

9-7. Principle of Operation of a Machine Tool with Finite Positioning Controls

The essence of systems of numerical controls will be explained using a Soviet-made machine tool as an example. The SKTBI Design Office, in conjunction with the Orjonikidze Machine Tool Plant, developed a series of N/C lathes with models 1712 and 1722 tracer-controlled semiautomatic lathes taken as the basic models. In any model of this series, the template can be replaced by a movable stop and a device for moving this stop and stopping it at the required points by means of a displacement transducer and in accordance with the given programme. If, in addition, a displacement transducer is incorporated to check the longitudinal position of the carriage, the machine is converted into a templateless general-purpose semiautomatic lathe, i.e., an N/C machine tool.

Let us assume that this lathe is to machine a stepped shaft with the dimensions specified in the drawing (Fig. 144*a*). The programmist selects one of the available machining cycles on the lathe, roughing or finishing, i.e., turning in several passes with return to the initial position, beginning with the largest diameter of step, or turning consecutive steps beginning with the smallest. His decision depends upon the shape of the workpiece. Next, he draws a processing drawing showing the consecutive passes, or compiles a corresponding table. Dimensions along the axis of the shaft should be specified from a single datum, the end face nearest the headstock. In this form, the data are given to the operator.

Mounted on the control panel are ten-position selector switches used for setting the dimensions for machining diameters and longitudinal dimensions of the workpiece. A separate row of switches is provided for each pass (Fig. 145). Without going into many details, the operation of the lathe, for instance on a roughing cycle, may be described as follows.

The tool (slide) is set in the initial position (in the upper right-hand position in reference to the blank). Upon pressing the control push button, motor M is switched on (Fig. 146), which drives stop-screw 1. This unit is mounted on the longitudinal carriage. The stop descends until it reaches a position corresponding to the first diameter to be turned. This will be noted by displacement transducer $DT1$. The stop ceases travel and, by means of a tracer and follow-up system (as in the conventional version of the hydraulic tracer-

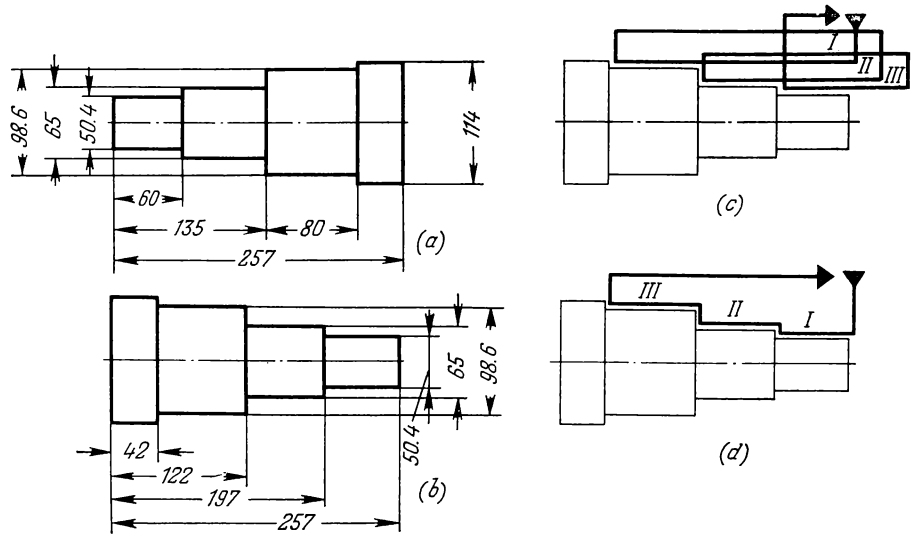


Fig. 144. Preparing the programme for turning a stepped shaft in the model 1712П semiautomatic:

(*a*) part drawing; (*b*) processing drawing; (*c*) and (*d*) roughing and finishing cycles, respectively (*I*, *II* and *III*—consecutive passes of the tool)

Passes	Diameter dimensions			Length dimensions			Cycle and processing commands
	×10	×1	×0.1	×100	×10	×1	
I	9	8	6	0	4	2	
II	6	5	0	1	2	2	
III	5	0	4	1	9	7	
IV							

Fig. 145. Panel for infeed of the programme in the model 1712П semiautomatic (shown schematically)

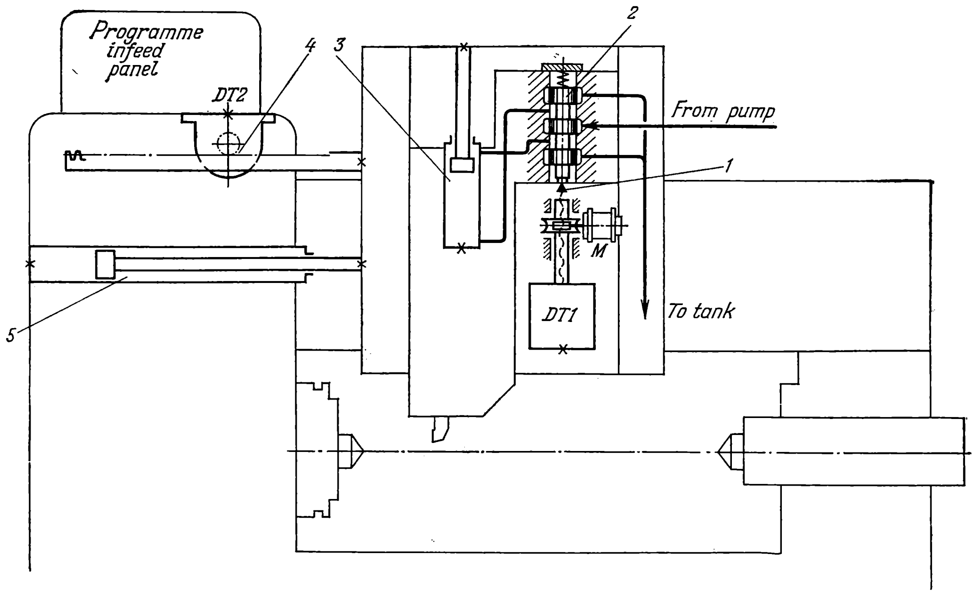


Fig. 146. Elementary diagram of the slide control system in the model 1712П semiautomatic:

1—stop-screw; 2—tracer valve; 3—hydraulic cylinder for cross feed of the tool; 4—rack-and-pinion drive; 5—hydraulic cylinder for longitudinal feed of the tool; M—electric motor with a worm reducing gear and a nut; DT1 and DT2—displacement transducers for cross and longitudinal travel, respectively

controlled lathe) the slide with the tool will take the working position for turning the first diameter. Next the drive is engaged for longitudinal travel of the carriage from the hydraulic cylinder. The carriage with the tool will be fed until the leading edge of the tool reaches the position set up on the switches for the first longitudinal pass. This position is checked by a second displacement transducer $DT2$. Then, the tool is automatically retracted from the work and returns to the initial position.

The next pass begins when the selector in the control system goes over one step, cutting in the next row of ten-position switches and cutting out the first row. The switches of the second row store the programme for machining the workpiece on the second pass. The tool travels in reference to the blank until it reaches the position corresponding to the turning of the second diameter. Then it is fed along the shaft, turning the next step, etc. All in all, the lathe can turn nine different diameters with nine different lengths.

Figure 147 shows a simplified electrical circuit diagram of the controls for the crosswise motions of the slide (tool). This circuit is based on the

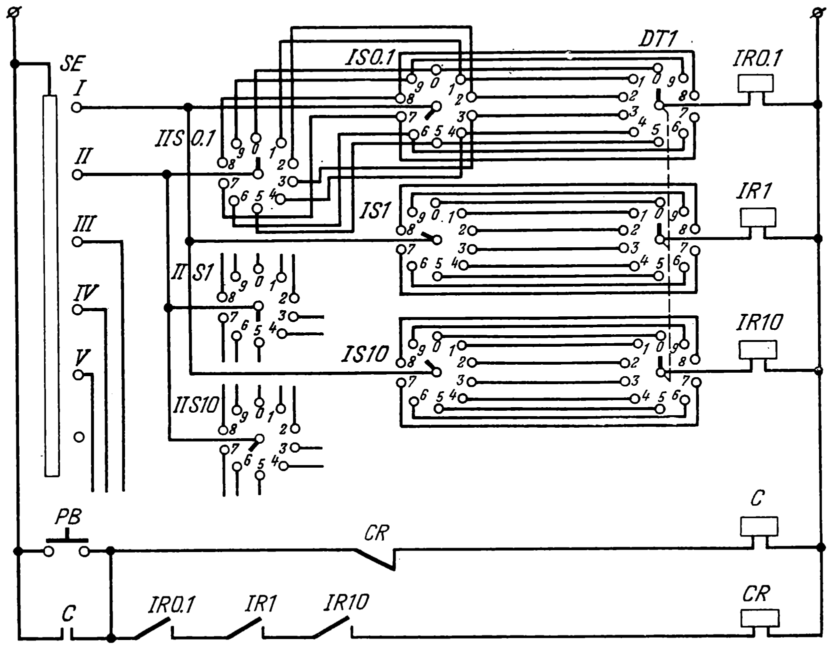


Fig. 147. Electric circuit diagram for the numerical control of cross travel of the tool in the model 1712П semiautomatic (simplified version):

PB—cycle start push button; *C*—contactor of electric motor *M* (see Fig. 146); *SE*—selector for the *I*, *II*, *III*, etc. positions; *IS*0.1, *IS*1, *IS*10, *IIS*0.1, etc.—ten-position selector switches of the control panel for the various orders and various passes; *DT*1—displacement transducer for cross travel of the tool with contacts for tenths, units and tens of millimetres; *IR*0.1, *IR*1 and *IR*10—intermediate relays for tenths, units and tens of millimetres; *CR*—coincidence relay

principle of coincidence of the output signal and the programmed signal, and is called a *coincidence circuit*. It operates as follows. Upon pressing the CYCLE push button *PB*, contactor *C* is closed, switching on motor *M* of the stop drive. The position of the stop is checked every 0.1 mm by displacement transducer *DT*1. This transducer is a device consisting of contact disks and brushes sliding along contacts. There are contacts corresponding to tens, units and tenths of a millimetre. In construction, the transducer resembles an ordinary decimal counter. Upon rotation of the brush for tenths of a millimetre, during which time it passes over ten contacts, the brush for whole millimetres moves up one contact. When the brush for whole millimetres makes one revolution (passing over ten contacts) the brush for tens of millimetres moves up one contact (the construction is taken up in more detail in Chapter 13). If the stop is set to a position in which the cutting

edge of the tool is on the line of centres, all the brushes of the transducer stand on contacts corresponding to the zero values, i.e., they indicate the number 00.0. As the stop moves away from the line of centres, the brushes of the transducer pass from contact to contact. This continues until they reach the contacts that were connected to the control circuit beforehand by setting the switches *IS* on the control panel according to the programme. Only if relays *IR*0.1, *IR*1, and *IR*10 are closed at the same time, will the coincidence relay *CR* be closed. As the normally closed contact of *CR* is opened, the holding circuit of the contactor *C* is broken and motor *M* is switched off. At this the stop occupies the position corresponding to the number set up on the switches for turning the first diameter.

The surface of the next diameter (second operation) is machined automatically (not shown in the diagram). The necessary information is fed into the control circuit by the selector *SE*, which connects switches *IIS* to the circuit of transducer *DT*1 in place of switches *IS*. A similar circuit controls longitudinal travel of the tool.

CHAPTER 10

PROGRAMMING AND METHODS OF CODING INFORMATION

10-1. Programming. Systems of Notation and Codes

The preparation of information for machining workpieces—*programming*—consists in transforming the processing and geometrical characteristics of the part (specified by the drawing) into a series of commands and signals representing the displacements of the operative units of the machine tool and in recording them in some arbitrary code on the storage medium or entering them into the machine by means of selector switches. Programming is a tedious process, especially in machining three-dimensional surfaces of complex shape. For this reason, in some cases, the mathematical calculations in converting the numerical information are done with the aid of quick-acting electronic computers. In simple cases, the programme can be computed manually, using key-type calculators or adding machines. The data for finite positioning systems are calculated either manually or by a computer. The resulting data are entered into the machine by setting switches or are recorded on a storage medium in the form of various combinations of holes punched in cards or tapes. Each combination of holes corresponds to a number which represents the specified displacement of the operative unit of the machine tool.

Various methods of coding numbers and various systems of notation are employed to record information. The most widely used are the following:

Decimal system of notation. The ordinary system of notation is a mark-position system based on ten different marks (characters)—the digits 0, 1, 2, 3, 4, 5, 6, 7, 8 and 9. Numbers greater than 9 are formed by replacing the last of the admissible marks of the system (9) by the first admissible mark of the system (0), while the mark in the next place to the left is replaced by the next larger mark (1 in our case). Thus, numbers from 10 through 99 are written by using two digits; numbers from 100 through 999 by using three digits, etc.

Since we are used to the decimal code, we can easily read it from holes punched in the storage medium (each hole meaning one of ten digits in the corresponding position). Such a code requires considerable space, being uneconomical in this respect. For example, to record any three-digit number from 000 through 999 by means of holes, thirty free positions are required (three places of ten digits each) of which three are punched for each number.

This system, being the most easily understood one, is recommended when the programme is entered by means of selector switches mounted on the control panel of the machine tool, or for feeding the programme into the perforator.

Binary system of notation. This system is based on the same law as the decimal system, but has only two admissible marks (two digits)—one (1) and zero (0). The values of each binary position (or place) beginning from the last one are: $2^0 = 1$; $2^1 = 2$; $2^2 = 4$; $2^3 = 8$; $2^4 = 16$; etc. Consequently, the number 26, for instance, is written in this system as 11010 ($= 1 \times 2^4 + 1 \times 2^3 + 0 \times 2^2 + 1 \times 2^1 + 0 \times 2^0 = 16 + 8 + 2$). The advantageous feature of the binary system is that each position (or place) can be expressed, not with ten marks as in the decimal system, but with two. This can be accomplished by having the physical elements in the circuit assume any two distinct states of being, for example: 1 ("one")—current in the circuit, and 0 ("zero")—no current in the circuit. For this any apparatus with two fixed positions will do, such as an electromagnetic relay (closed—1, open—0) or holes in a punched card (hole—1, no hole—0). Only ten free positions are required to record any decimal number from 0000 to 1023 on the storage medium (since $2^{10} = 1024$). Any number of positions, from 0 to all 10, may be punched.

This type of coding is the most economical; its application requires a storage medium of minimum capacity and small number of communication channels and apparatus.

Some of the rules for handling binary numbers are:

1. In adding binary numbers, $0 + 0 = 0$; $0 + 1 = 1$, and $1 + 1 = 10$.

2. To convert a decimal number into a binary number divide the decimal number by two. If a whole number is obtained, write 0 in the lowest position (place) of the binary number (at the right). If there is a remainder of one, then 1 is written in this position. Next the quotient obtained in the division (neglecting the remainder) is again divided by two. The procedure is the same as before except that the 0 or 1 is written in the next higher position (to the left). This continues until the quotient is equal to unity which is written down in the highest position (at the extreme left). In practice, the numbers are written in two columns. Arranged at the right are the remainders obtained in each division (0 or 1), which, if read upward, give the binary number. For example, to convert 26 in a binary number we write:

26	0
13	1
6	0
3	1
1	1

reading upward: $26 \rightarrow 11010$

It is easier to convert large numbers by writing them as follows for the number 214:

$$214^0 107^1 53^1 26^0 13^1 6^0 3^1 1^1, \text{ i.e.,}$$
$$214 \rightarrow 11010110$$

3. To convert binary notation into decimal notation it is necessary to know what the "one" in each position of the binary number is equal to in the decimal system (what the value of each "one" is). Then, in conversion to the decimal system, all the values of the binary positions, expressed in the decimal system, are added together. If in any position of the binary number we have a 0, this zero is retained in conversion to the decimal system. Thus: $11010 \rightarrow 16 + 8 + 0 + 2 + 0 = 26$, and $11010110 = 1 \times 2^7 + 1 \times 2^6 + + 0 \times 2^5 + 1 \times 2^4 + 0 \times 2^3 + 1 \times 2^2 + 1 \times 2^1 + 0 \times 2^0 = 128 + 64 + 0 + 16 + 0 + 4 + 2 + 0 = 214$.

Relay circuits, such as the "pyramidal circuit" (Fig. 148) and the diode matrix (Fig. 149) are used for automatic conversion from the binary to the decimal system and back again. These circuits can be employed in *encoders* and *decoders*, i.e., in devices for transforming information. A *diode matrix* is made up of horizontal and vertical busbars. Depending upon the required conversion, the busbars are connected at the necessary points of intersection, either directly or through a semiconductor diode. Connection by means of a diode (gate-type, i.e., unidirectional) has the purpose of excluding the appearance of spurious signals at the output.

Tally system of notation. Only one mark (symbol)—figure 1—is used in this system. Hence, to convert any decimal number, for example 26, into tally notation, it is necessary to write the figure 1 twenty-six times in a row. This system is very convenient in transmitting information along communication channels since each signal pulse carries one unit of elementary displacement (it is accepted practice to speak of a definite value of the pulse). Such a system of encoding is called a *unitary* or *tally code.* The system is very unwieldy in recording and is not used with a punched storage medium. The system requires as many positions as there are units in the given number.

The tally code is widely used for recording information on magnetic tape whose specific capacity (per unit of length) is many times greater than that of punched storage media.

Tally-coded decimal encoding is obtained when each position, or place, of the decimal system is dealt with in the tally system of notation. This procedure retains the advantage of the tally system, in that one mark characterizes a quite definite displacement, equal to the elementary displacement or a multiple of 10 (in this system the value of the pulses will differ for the various positions). At the same time less space is required for recording than in the tally system. The figures of the positions of the decimal number are recorded by perforating, the number of punched holes being equal to the

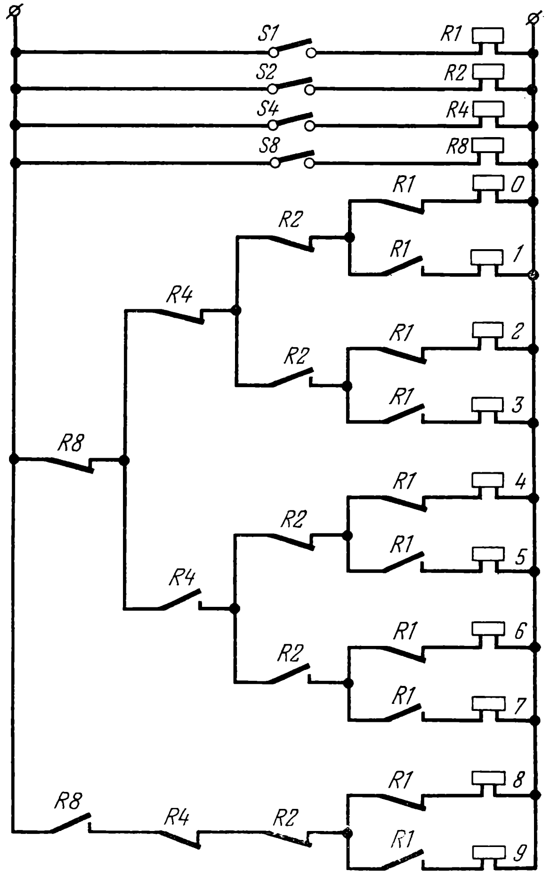


Fig. 148. Circuit diagram of a relay decoder used for converting the binary system of notation into the decimal system:
S1, S2, S4, S8, R1, R2, R4, and R8—switches and relays for entering binary-coded information

figures. The positions are separated from one another by recording along different paths on the storage medium.

Binary-coded decimal system (*8421 code*). When this system is used for encoding, each digit of a number given in the decimal system is represented separately in binary notation. Thus, 214 → 10 | 1 | 100.

To obtain symmetry in recording, zeros are added in front of each binary number so that they are each four places (bits) long. The final form of the

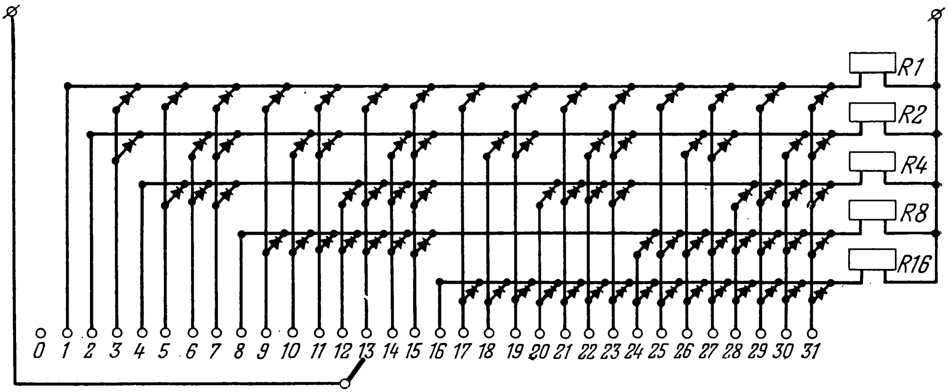


Fig. 149. Circuit diagram for converting numerical information from the decimal system of notation into the binary system by means of a diode connection matrix

notation for the number 214 is 0010 | 0001 | 0100. Though this system is somewhat less economical in respect to the required space than the straight binary system (for instance, twelve free positions are required to record a three-bit decimal number), it is easier to comprehend since the coded number can be read and translated more easily into a decimal number (Fig. 150).

Coded decimal systems (*5211*, *4311*, *4221 and 3321 codes*). These codes differ from the ordinary binary-coded decimal system (8421 code) in that here the higher positions have different values and in that the sum of all the positions is equal to 9 instead of 15. The choice of a code depends upon the conditions stipulated for the design of the various units of a numerical control system. If the specific features of the circuit are taken into account, as well as the probability that certain figures will appear in the programme, and the most frequently encountered figures are expressed with the minimum number of symbols (and, consequently, with the minimum number of elements for reproducing information), while infrequently encountered figures are expressed with the maximum number of symbols, a so-called *optimal code* is compiled. For some cases, the 2421 and 5121 codes are optimal (as are the 3321 and 4221 codes for arrangements with measuring stops).

Special codes. These codes are not summating, i. e., their positions (or places) do not have a definite value, and each decimal number corresponds to a definite encoded combination of signs. Thus, the two-out-of-five code is a combination of five symbols taken in twos.

If the programme is read automatically, a modified binary-coded decimal system is used to check the information on whether the sum of the 1's of the encoded combinations is odd or even. An additional sign (position) is provided

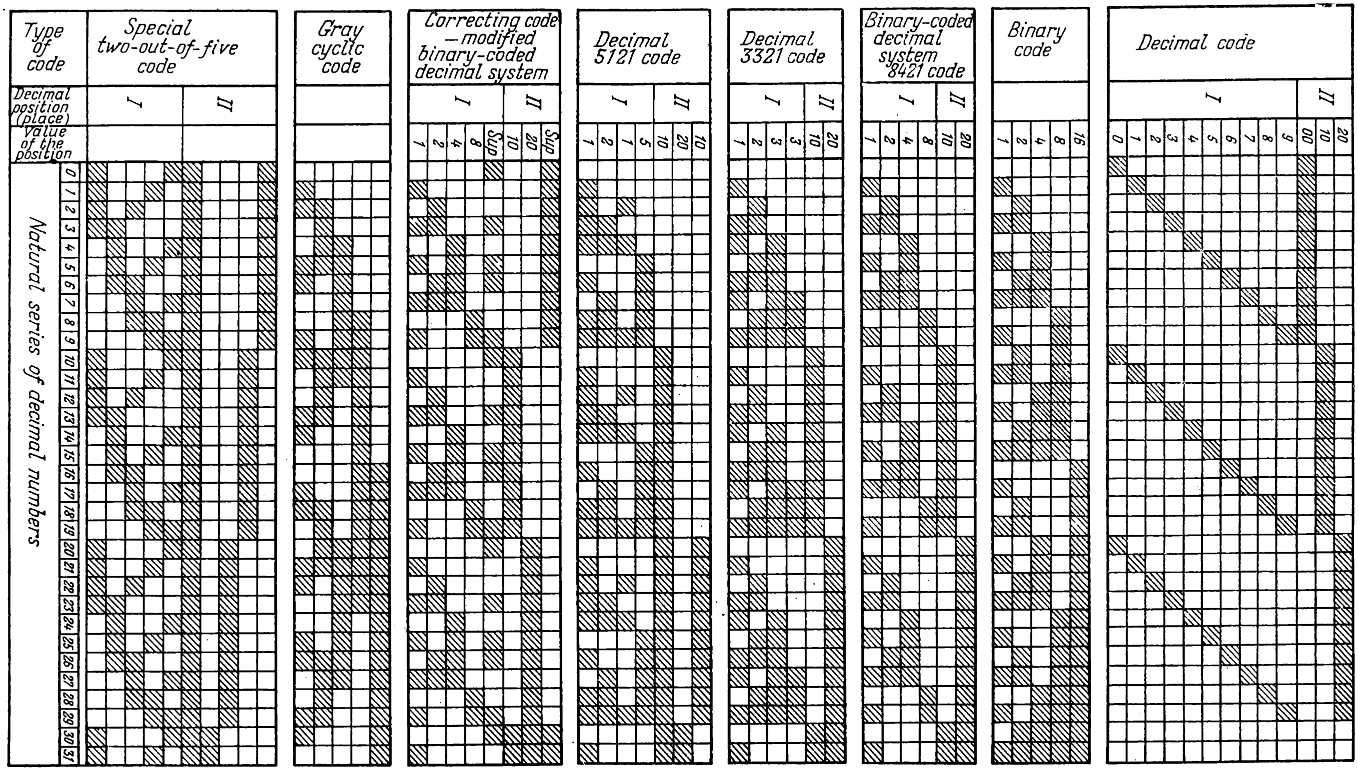


Fig. 150. Certain methods of encoding numerical information

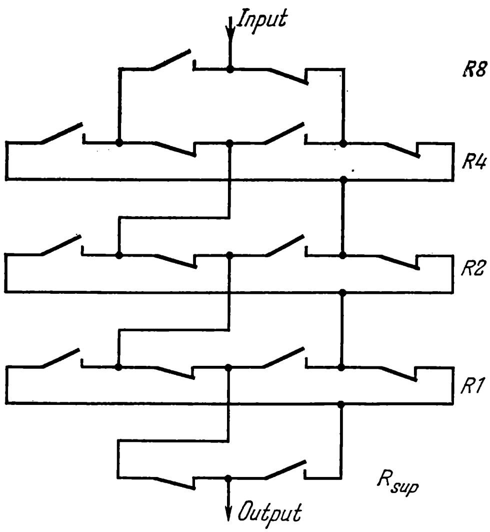


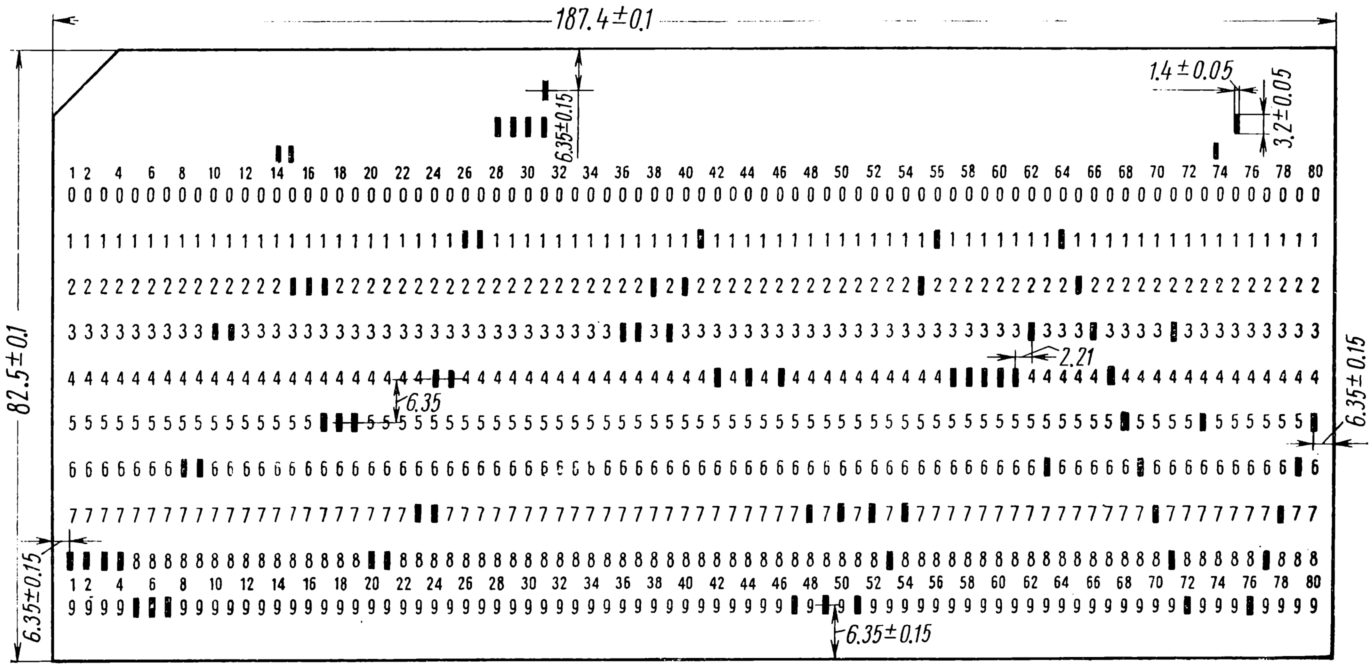
Fig. 151. An example of a circuit for checking numerical information in the modified binary-coded decimal system of notation for the oddness of the number of signals of code combinations (the coils of the corresponding relays are not shown):
R_{sup}—contacts of a supplementary relay participating in checking oddness (if the normal position of the contacts is changed to its opposite, the check will be for an even number of signals)

for this purpose. Such *correcting codes* are *redundant* in respect to the numerical information they represent; some of the code symbols are used for transmitting information and the remainder are used for checking purposes. A special circuit (Fig. 151) is used to check whether the information is being properly recorded and read. This circuit passes a signal only if the sum of the digits of the code combination is odd—1 or 3. This reduces the possibilities of errors occurring but does not, of course, completely exclude them.

A modification of the binary code is the *Gray code* which is one of the cyclic binary codes. To obtain the Gray code, it is necessary to form new combinations from the combinations of the binary system, eliminating the signs of the lowest position and moving all the signs one position to the right. These combinations are added, position to position (without carrying over the 1's to the higher position) with the combinations of the binary system (Table 5).

The application of cyclic systems of encoding is one of the methods of eliminating spurious codes (signals) (see p. 238).

The systems of encoding have been standardized in some countries.



(a)

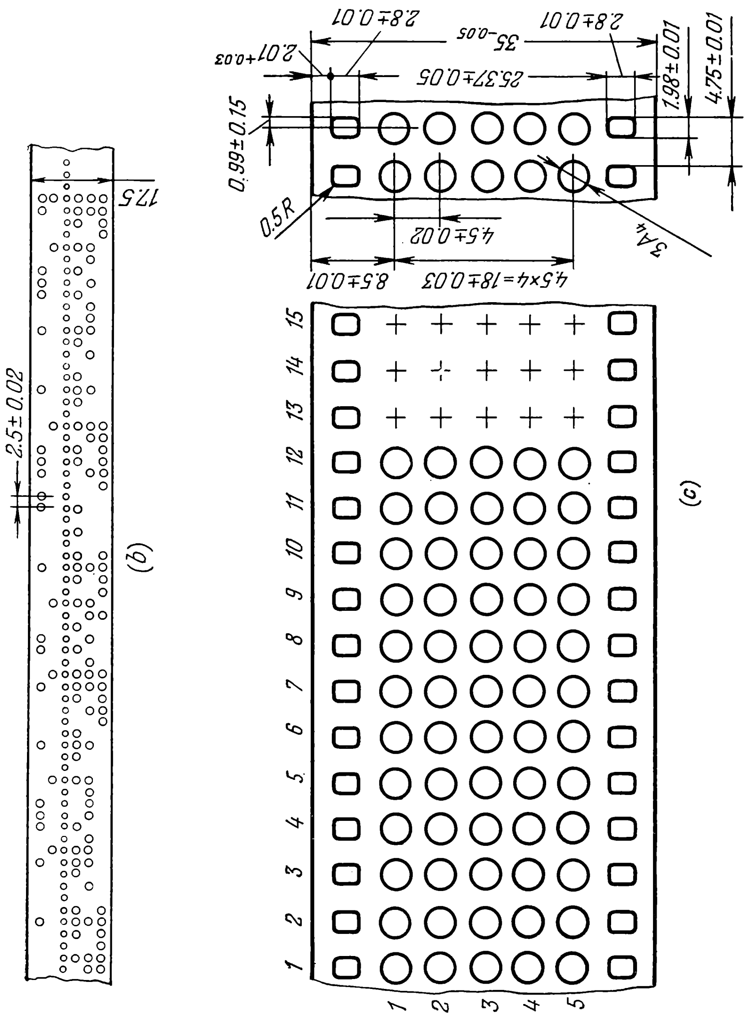


Fig. 152. Types of punched storage media:
(*a*) eighty-column punched card; (*b*) teletype punched tape; (*c*) 35-mm punched tape

TABLE 5

Natural series of numbers	Binary code	New combination	Cyclic binary code	Natural series of numbers	Binary code	New combination	Cyclic binary code
0	0000	0000	0000	5	0101	0010	0111
1	0001	0000	0001	6	0110	0011	0101
2	0010	0001	0011	7	0111	0011	0100
3	0011	0001	0010	8	1000	0100	1100
4	0100	0010	0110	9	1001	0100	1101

10-2. Storage Media and Perforators

As mentioned previously (in respect to the semiautomatics, models 1712П and 1722П) information on the machining of a workpiece is fixed by setting selector switches on the control panel of the machine tool. This simple method of entering the programme has an essential shortcoming—the programme cannot be stored and subsequently repeated. To repeat the operation, it is necessary to set all the programmed dimensions and commands on the selector switches again. A more economical procedure is to have a cheap interchangeable storage medium which can be conveniently stored. Punched cards or tape are commonly used for this purpose (Fig. 152). Information is recorded on the cards or tape by means of a perforator which punches holes in them in a definite order according to the selected code.

In the USSR, standard paper punched cards of the Soyuzmashuchet type (USSR Std GOST 6198-64) have found application for machine tools with finite positioning controls. These cards are available with 45 or 80 columns and with 12 rows. Thus, information can be recorded in any of 960 positions in an 80-column card. Recording may be carried out either along the columns or along the rows. Punched tape is used to store a larger amount of information. Used for this purpose is plastic tape (similar to cinema film) 35 mm wide (according to USSR Std GOST 4896-49) or paper tape 17.5 mm wide (USSR Std GOST 1391-51) used in telegraphic apparatus. Information can be recorded on the tape in lines (across the tape) or in blocks of definite area.

Hand perforators of the paper-puncher type (Fig. 153) may be used in addition to keyboard perforators with a built-in encoder for programme recording.

Two different systems of perforators are available: single- and two-stage. In the single-stage perforators, simultaneously with the selection of the data, effected by pressing the keys, the holes are punched in the card or tape cor-

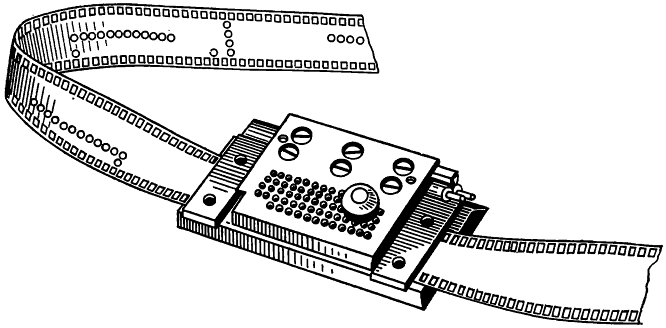


Fig. 153. Hand perforator

responding to the separate signs or symbols. In the two-stage perforators, first the information (numbers and commands) is set up. At the end of this stage all the holes are punched simultaneously. The two-stage perforator of the Soviet electronic computer, model M-20, can be used for row-by-row recording of information on an eighty-column punched card. This perforator has a high output and can be recommended for application in united programming centres. The model П-80 perforator (Fig. 154) can be recommended for punching the same cards along the columns. The model П-2 single-stage perforator (Fig. 155), designed in ENIMS, can be advisably employed for punching 35-mm tape in lines, while the model CT-35 printing telegraph set (teletype) can be used for punching paper tape 17.5 mm wide. A perforator, developed by Friden (USA) for use in machine tool control systems, has very favourable features (Fig. 156).

Various methods can be employed to check the information recorded on the storage medium—from primitive visual inspection to automatic inspection performed by special equipment. In using punched cards it is advisable to punch two identical cards at different times and to check whether the holes have been properly punched by laying one card on the other. There is little probability of the repetition of the same error when two cards are punched independently.

In the Friden perforator, simultaneously with the punching of the tape, information in the form of numerical and textual data is typed on a sheet of paper. In checking the recorded programme in the same perforator, the punched tape is passed through again. This punches a duplicate tape and types the recorded programme again. After this the two texts are compared visually and checked against the initial tables.

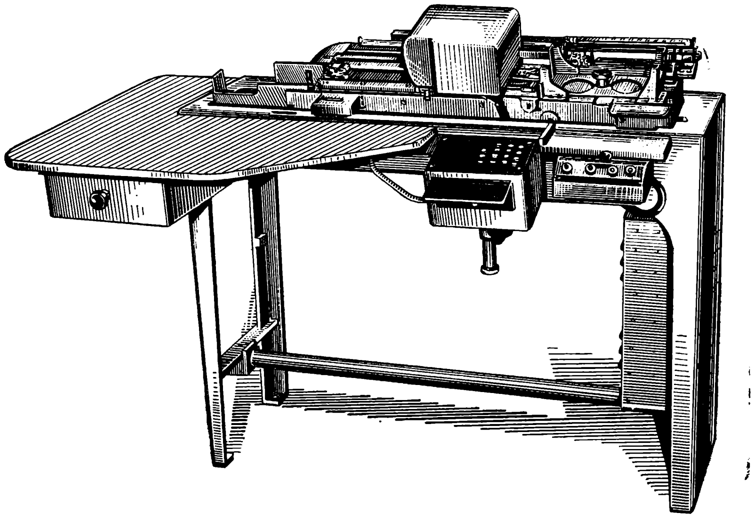


Fig. 154. Perforator, model П-80

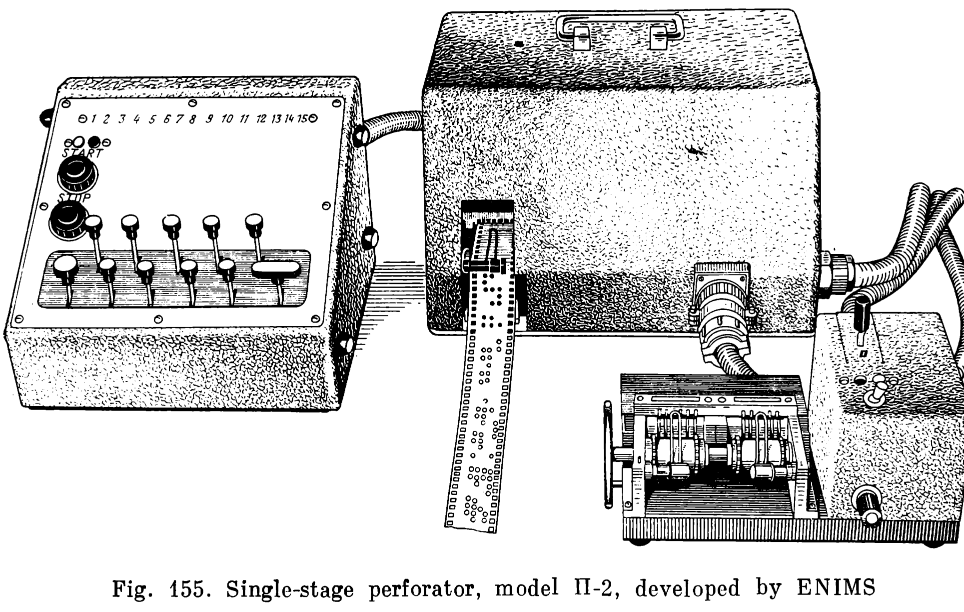


Fig. 155. Single-stage perforator, model П-2, developed by ENIMS

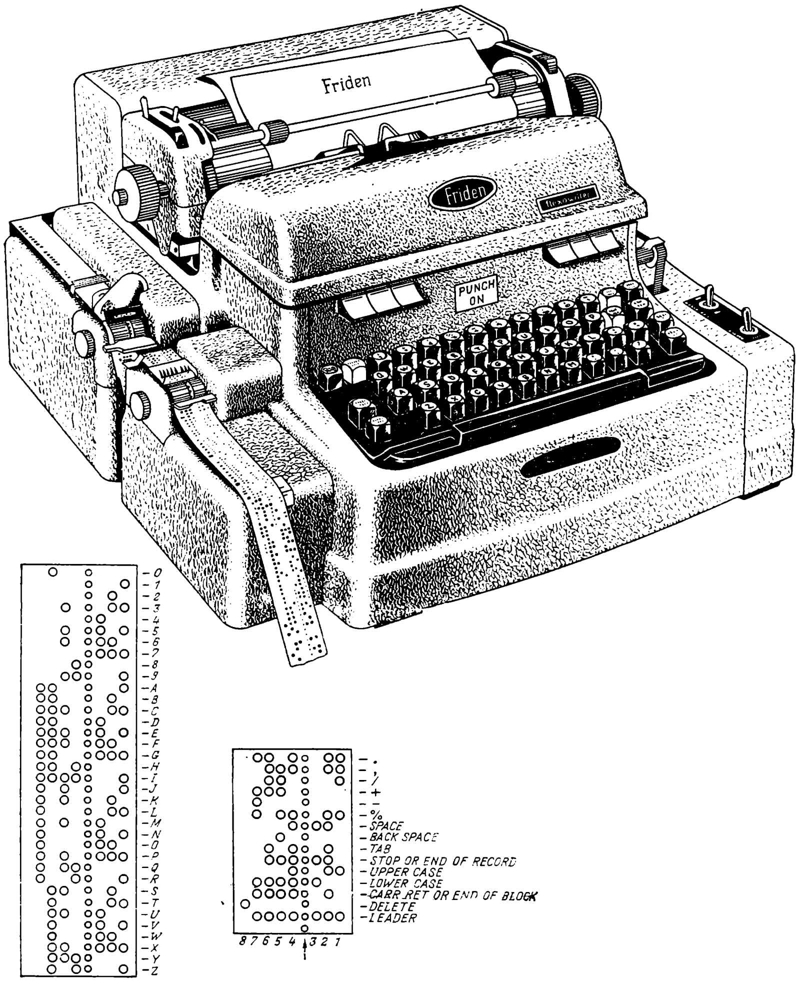


Fig. 156. The Friden perforator (USA) for 8-path punched tape 1″ wide and the code used on this tape

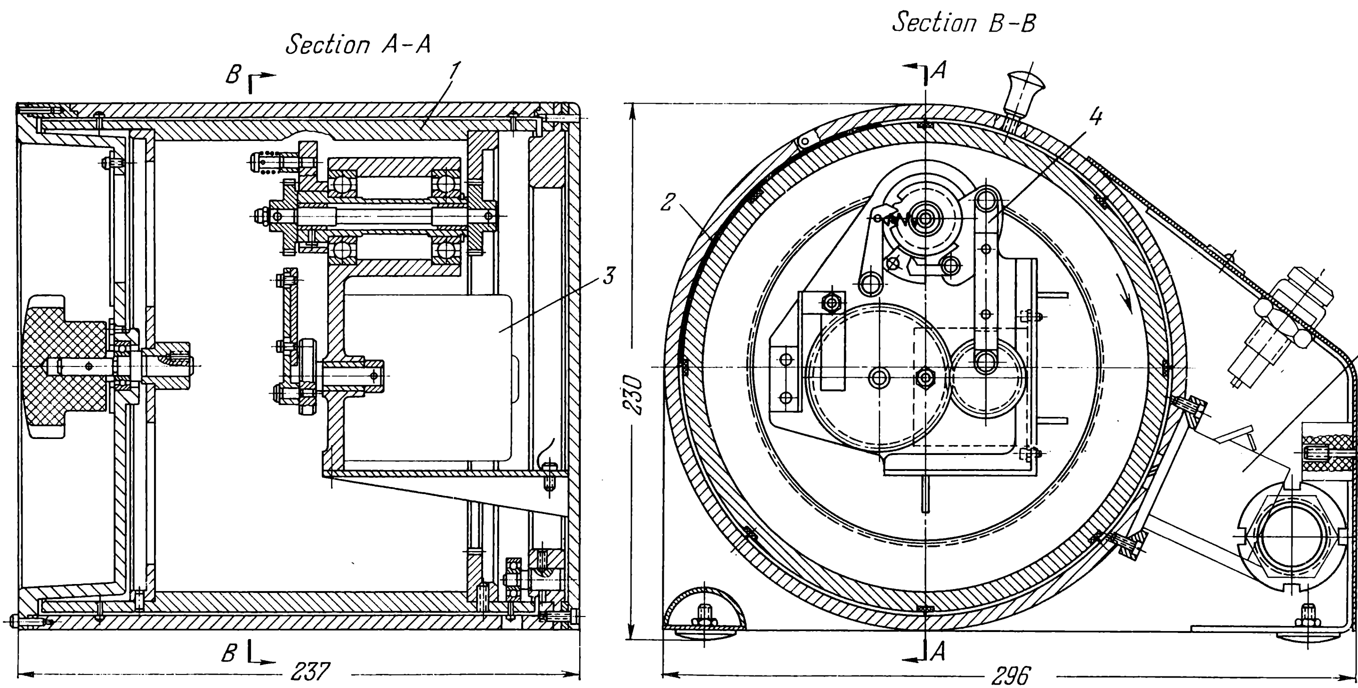


Fig. 157. Rotary device for reading punched cards:
1—drum; 2—punched card; 3—electric motor, type РД-09; 4—ratchet mechanism; 5—brush contact reader unit

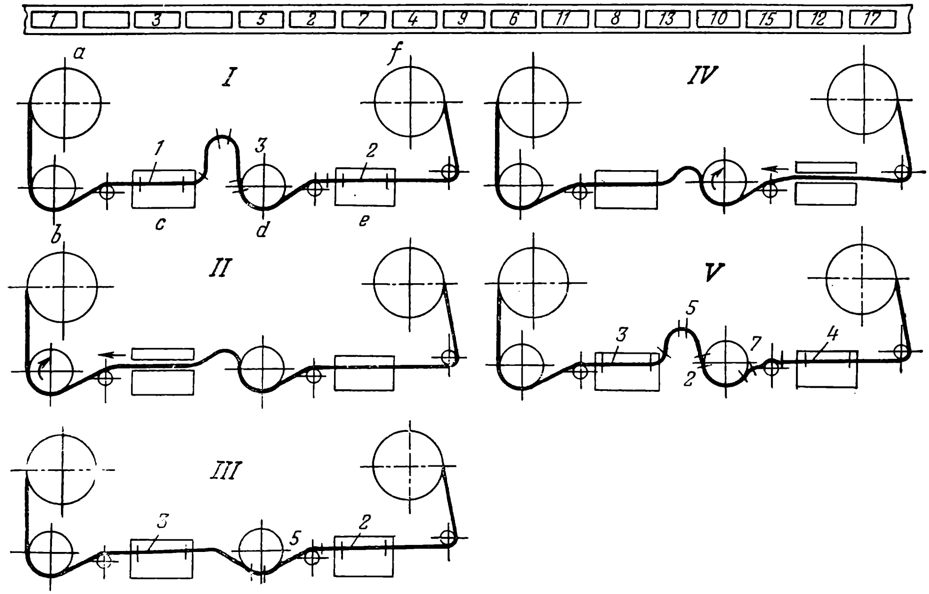


Fig. 158. Principle of the device for continuous reading of the programme (developed by ENIMS):

Above—order of the frames in the punched tape; *I*, *II*, *III*, *IV* and *V*—consecutive stages in the operation of the device; *a* and *f*—take-up and feed tape reels; *b* and *d*—left- and right-hand sprockets (engaged alternately); *c* and *e*—left- and right-hand tape-reading units (reading alternately)

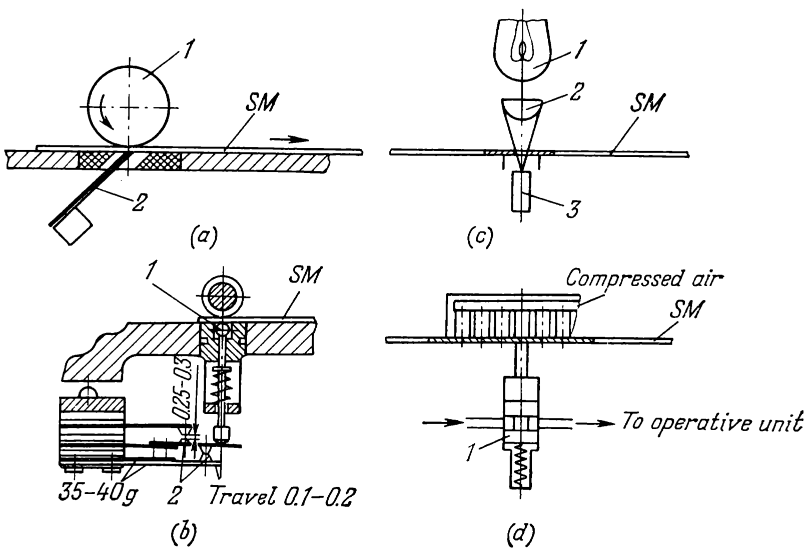


Fig. 159. Various methods of reading information from a punched storage medium:

(*a*) electrical (*1*—contact drum; *2*—contact brush); (*b*) electromechanical (*1*—feeler ball; *2*—contact unit); (c) photoelectric (*1*—electric lamp; *2*—lens; *3*—photocell); (*d*) pneumatic (*1*—valve); *SM*—storage medium

10-3. Devices for Feeding the Storage Medium and for Reading the Programme

Depending upon the type of storage medium employed, special reading and handling devices are used for punched cards and tape feeding mechanisms for punched tape. Punched cards are used for a comparatively small amount of information, usually in cases when several dozen cards are sufficient for the operation of the machine tool during a shift. Therefore, a rotary (drum) card feeder with contact reading facilities is sometimes used. A design developed by a Soviet jig-boring machine plant is given as an example (Fig. 157). Up to eight punched cards are arranged crosswise on the stainless steel drum. They are held by teeth of the drum which fit into the feeding holes located along the sides of the cards. Driven by an electric motor through a ratchet mechanism, the drum is indexed one pitch, equal to the distance between the rows on a punched card, during one cycle. Reading is accomplished by a standard brush contact reader unit from an eighty-column tabulator.

Designs, adapted from book-keeping machines, have a card hopper and a device which feeds the card out of the hopper and brings it, row after row, consecutively, to the reader.

Storage units are incorporated in the designs of N/C machine tools operating by contouring control. They retain the information during the time the tape is being fed to a new position for reading. In some cases the programme reader has two alternately operating tape-reading units, as in a design developed in ENIMS. If the programme is to be read in this way, it is recorded, by punching the tape, in a certain definite order, namely: all even blocks are shifted two places (four blocks) to the right and the recording is as shown in Fig. 158. In this case, the reading units read out the blocks recorded on the punched tape, not in succession, but in the sequence of the information, i.e., in the order the blocks are numbered. To accomplish this, the programme reader feeds part of the tape between the units while the left-hand unit is reading an odd block and prepares a new even block for reading by the right-hand unit. As the right-hand unit reads this block, the reader feeds the tape again between the units preparing a new odd block for reading, etc.

Shown in Fig. 159 are different methods of reading a programme recorded on a punched card or tape. Each method has its advantages and shortcomings. Evidently, at high rates of information reading, noncontact reading devices should be used.

CHAPTER 11

METHODS OF CHECKING THE POSITION AND DISPLACEMENT OF MACHINE TOOL OPERATIVE UNITS

11-1. Open- and Closed-Loop Control Systems

Most machine tools have operative (movable) units which participate either in the motions for shaping the workpiece (by accomplishing working travel movements) or in positioning (idle) motions of the workpiece or cutting tool by performing approach and withdrawal movements or co-ordinate positioning. Examples of such units are the carriage of a lathe or the table of a boring, drilling or milling machine. It has been previously stated that in general-purpose machine tools, the operator controls and checks the motions of such units, using for this purpose visual readings that he makes on the measuring or reading facilities with which such machine tools are equipped. In automatic and semiautomatic machine tools, this is accomplished automatically by means of cams mounted on a camshaft or by trip dogs and limit switches or, finally, by means of templates and cam bars in tracer-controlled machines. In N/C machine tools, the methods employed to check the positions of the operative units can form the basis for the classification of these machine tools in accordance with their design.

It was mentioned in Sec. 9-4 that N/C systems may be *closed-looped* (beginning with those having only simple internal linkages and up to systems completely covered by feedback) with various position transducers, or they may be *open-looped*. In open-loop systems, the operative unit should have a drive of a type enabling the path travelled by the unit to be checked without resorting to a transducer. This can be accomplished both in finite positioning and in contouring systems if the operative unit is displaced in steps that are constant in magnitude (metered) and of a controlled quantity. Examples of mechanical drives of this kind can be drives with a ratchet mechanism, Geneva wheel or a single-revolution clutch. In N/C systems, an electric drive in the form of electric step motors is commonly employed (see p. 269). Depending upon the force required to traverse the operative unit, the electric step motor can be either directly linked to the operative unit (constituting a power drive), or it can be a servomotor operating in conjunction with suitable amplifiers (see Fig. 160).

The use of position transducers is characteristic of closed-loop N/C systems. In these cases, the operative unit drive may have a wide range of variation according to the ordinary servosystem circuit with feedback accomplished

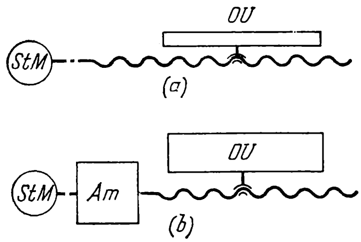


Fig. 160. Open-loop control systems with step drives:
(*a*) with a power step motor (*StM*); *OU*—operative unit; (*b*) with a step motor as a servodrive and an amplifier (*Am*)

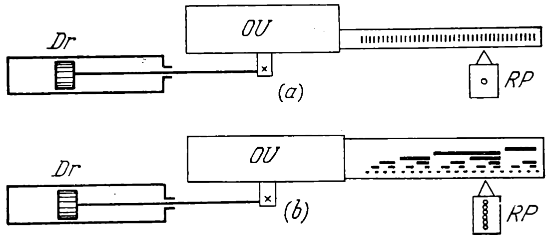


Fig. 161. Examples of closed-loop control systems:
(*a*) with a pulse-type transducer, the circuit being based on the counting principle; (*b*) with an encoded transducer, the circuit being based on the reading principle (coincidence circuit); *Dr*—drive of the operative unit *OU*; *RP*—transducer, consisting of a pulse (circuit *a*) or encoded (circuit *b*) scale and a sensing member for reading the scale

by a transducer (mainly in contouring systems), or it may work according to a simpler circuit (only used in finite positioning systems) providing for stopping the operative unit (with permissible accuracy) at the point specified by the programme from commands transmitted by the transducer. Such a stop can be obtained by intensive braking, by disengaging the masses having the highest inertia torques or by some other method of acting on the operative unit. In such cases, it would be more correct to speak of automatic control systems with spurious feedback, since the correcting signal is not generated and the operative unit may have free uncontrolled overtravel. The control system resembles an in-travel control circuit with contacting or noncontacting limit switches which are automatically "disposed" by the numerical programme along the line of travel of the operative unit with the corresponding accuracy and discrete features (in accordance with the resolving capacity of the given system).

Transducers or measuring elements of finite positioning systems, or more exactly, their scales, can be designed either on the principle of counting the divisions of the measuring scale, or on the principle of the reading of the scale (Fig. 161). In the second case, they operate in so-called coincidence circuits (see p. 211).

Such transducers are usually of the *discrete* type, i.e., their information is fed into the control system intermittently, no information being given out by the pickup in the intervals between the transmission of signals, and the travel of the operative unit along its ways being uncontrolled in these intervals.

In contouring systems, the transducers are designed for use in servosystems of either the pulse or phase type.

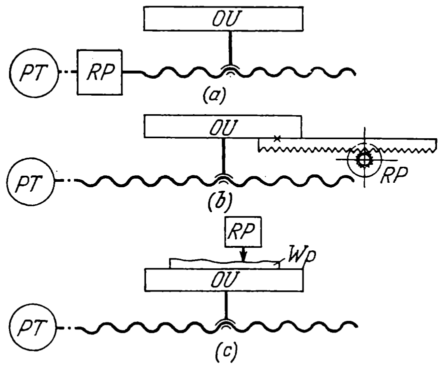


Fig. 162. Possible methods of installing the transducer for various degrees of coverage of the system with feedback (more or less closed-loop features):

(*a*) transducer (*RP*) in the power train (*PT*) of the drive; (*b*) transducer with an independent drive from the operative unit (*OU*); (*c*) transducer effecting automatic sizing of the workpiece (*Wp*) with feedback controls

A phase-type control system, for example one using selsyns or rotary resolvers as feedback transducers, or a system with voltage control, belongs to the analog, i.e., continuous control, systems, and transducers from which information is being continually transmitted are called analog transducers. They can be used both in finite positioning and in contouring systems. It should be noted, however, that programming in N/C machine tools is always of a discrete nature, since it is accomplished by means of numbers.

Transducers can be installed in the power drive train or they can have an independent drive from the operative units, or they can be rigidly linked to these units (Fig. 162). The accuracy of a system with an independent drive or of one in which the transducer is rigidly linked to the operative unit can be improved by greater coverage of the system with feedback (more closed-loop features), and the absence of loads leading to deformation and measuring errors in the transducer drive train. In this respect, the best systems are those with direct checking of the shape and size of the workpiece, i.e., N/C systems having automatic sizing with feedback controls (Fig. 162*c*). Such systems are not applied as yet due to their complexity. In machine tools with finite positioning controls, the position of the operative units is checked in some cases by a system which uses adjustable sliding stops (with mechanical, hydraulic or electric drives) or magnetic lines recorded on tape during the time the first workpiece of a batch is being made with manual control. An electromagnetic stop patented in Czechoslovakia (Fig. 163) is of interest. Depending upon the numerical programme, various sections of the electromagnet, or combinations of them, are energized and the core (which is also the stop) is extended by the required amount. Thus with four sections having core travel values of 1, 2, 2 and 4 mm, ten positions of the stop are available (0, 1, 2, 3 . . . 8 and 9 mm).

Machine tools which are programmed by recording data obtained in making the first workpiece of a batch with manual controls are half way between N/C and semiautomatic machine tools; they have not found extensive application. The programming of the displacements of the operative units is quite flexible in these machines, but the measuring element, for instance,

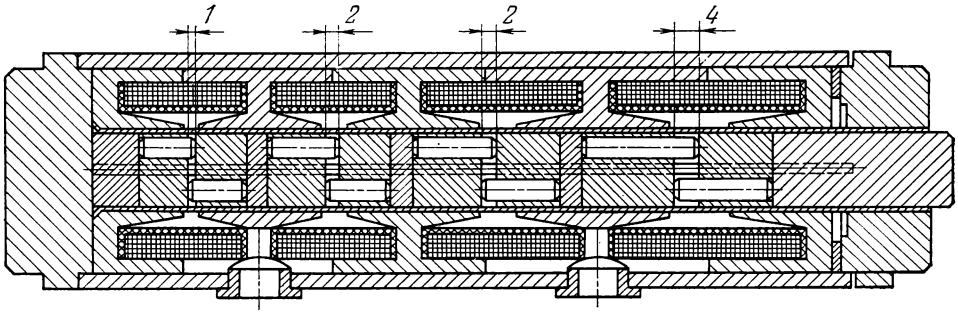


Fig. 163. Ten-position electromagnetic stop

a magnetic drum on which the magnetic lines are recorded during the machining of the first workpiece, is in fact a template, made to a certain scale, which is the physical materialization of the programme. Such storage media are inconvenient to handle and keep for further use. This system was employed in a previous model of an N/C jig borer made by Société Genevoise (SIP) of Switzerland. In a more recent model, the jig borer is controlled by information recorded on a punched tape. Recording during the machining of the first workpiece may prove expedient in N/C machine tools with contouring systems only in comparatively rare cases when it is too difficult to prepare the programme in any other way.

11-2. General Principles of Discrete Feedback Transducers

As previously mentioned, spatial values (positions of the operative units) are checked in closed-loop N/C systems by means of feedback transducers which, in a number of cases, are converters of spatial values into numerical form (i.e., into discrete numerical values). A device consisting of a linear or circular scale and a sensor ("reading" member) is called a *transducer* of the position or displacement of the operative unit (which we have called a response pickup for the general case). Transducers may be of the contacting or noncontacting type. The division of spatial scales—*quantization*—is done by using physical features associated with electrical, magnetic, optical, mechanical and other phenomena. Such features may be conductivity and nonconductivity, magnetization and nonmagnetization, transparency and opacity, projections and recesses, etc. An electrical scale can also be used; in this case, the measured values are converted into electrical analogs, for which purpose a selsyn or a rotary resolver is employed, for example, as the transducer. These are called analog transducers.

Scales are quantized by specifying either segments or discrete points (Fig. 164).

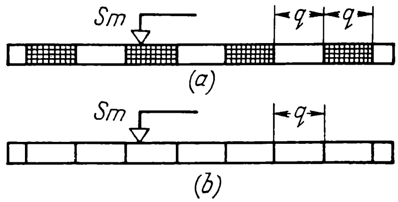


Fig. 164. Methods of quantizing scales: (*a*) specifying segments; (*b*) specifying discrete points; *Sm*—sensing member; *q*—elementary quantum of the scale

Spatial values can be encoded (i.e., represented in various forms) either on the principle of *counting the quanta* of the scale or by *reading the code* of the quantized scale. In encoding on the counting principle, the scale quanta are converted into electric pulses which are counted by special (as a rule, electronic) counters. The count may be made periodically (changing the measured value upon displacement of the operative unit), each time from a new zero point, or reversibly—from a common zero.

For the purpose of recognizing the direction of travel of the operative unit, and the scale or sensing member linked to it, in reversible counting with quantization by the specification of segments, two sensing members can be used, one being displaced in reference to the other by $\frac{1}{2}q$, where q is an elementary quantum of the scale (Fig. 165). Reversible counters are employed in these circuits. Upon travel of the scale from right to left, the recognizing device sends pulses through channel A and the counter adds them to the previously stored number. Upon reverse travel, pulses are sent through channel B and the counter subtracts them from the stored number.

The recognizing device is designed according to a logical circuit. Trigger *Tr1* forms signals from sensing member Sm_1 into two anti-phased square-waves which are transformed into two sequences of bipolar short-duration pulses by differential circuits a_1 and a_2. These pulses arrive at the first inputs of the logical "AND+" elements (b_1 and b_2). Each of these circuits has two inputs. The output positive voltage rises in a circuit if positive voltages act on both inputs. Trigger *Tr2* forms signals from sensing member Sm_2 into two phased square-waves biased by $\frac{1}{2}q$ in reference to the input signals of *Tr1*. These waves arrive at the second inputs of b_1 and b_2. Upon travel of the scale from right to left, a positive pulse rises on the first input when there is a positive potential from *Tr2* on the second input. In this condition, there are positive pulses on the output of b_1 (in channel A) and no pulses on the output of b_2 (in channel B). Upon reverse travel, coincidence of positive signals occurs on the inputs of circuit b_2 and output positive pulses appear in channel B. The pulses disappear at this time in channel A.

Encoding by the reading principle may be accomplished by two methods.

Encoding by means of a code combination field (Fig. 166). The field is built up for the binary system of notation as follows: quantized scales are arranged

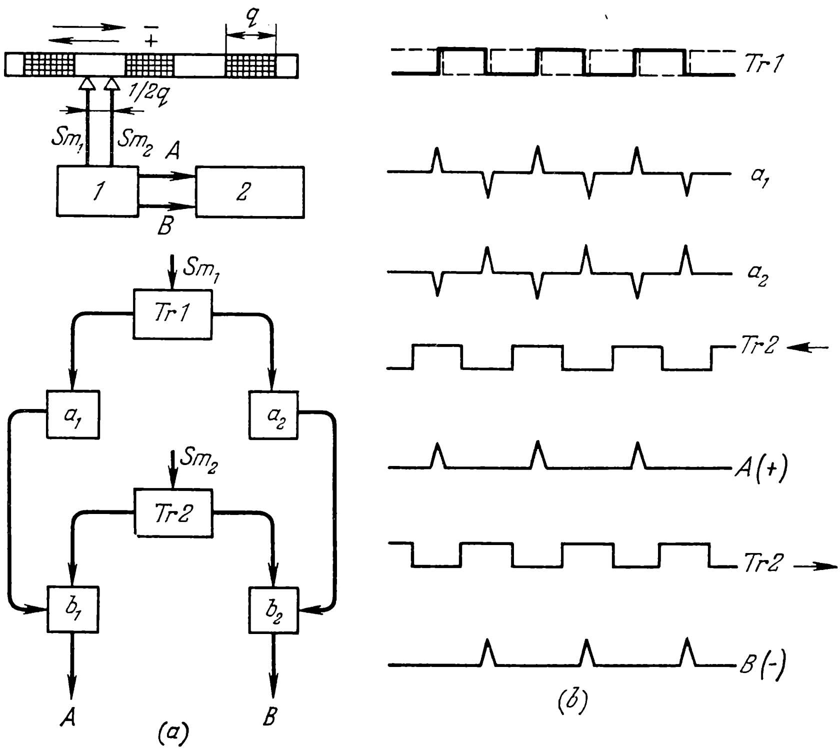


Fig. 165. Principle of a transducer operating with encoding by the reversible counting method (with a logical circuit for recognizing the direction of travel):

1—member for recognizing the direction of travel; 2—reversible counter; Sm_1 and Sm_2—sensing members spaced at a distance of $\frac{1}{2}$ quantum of scale q; A and B—channels for the output of signals for different directions of travel; (a) structural diagram of the recognizing member; $Tr1$ and $Tr2$—triggers; a_1 and a_2—differential circuits; b_1 and b_2—logical elements; (b) diagrams of signals passing through the recognizing member

parallel to one another so that their zero points coincide. The scales have differently valued quanta (divisions corresponding to the positions or places of the number). Thus, $q_0 = q$, $q_1 = 2q$, $q_2 = 4q$, $q_3 = 8q$, etc. Upon travel of the code combination field in reference to the sensing members (for instance, brushes), or vice versa, these members "read" the codes of their positions (places).

Encoding by converting the scale quanta into code combinations. The conversion of the natural series of numbers into binary code combinations by means of a connection matrix is illustrated in Fig. 167. A drawback of matrix

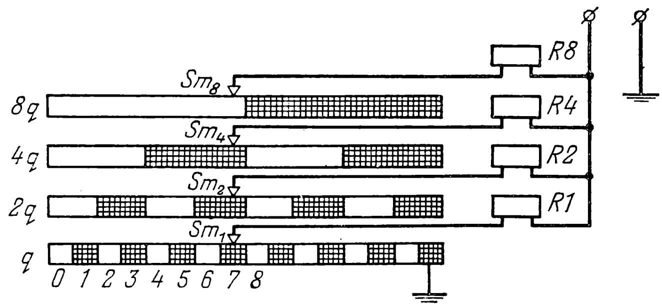


Fig. 166. Encoding a scale in the binary system of notation by means of a code combination field:

Sm_1, Sm_2, Sm_4, Sm_8, $R1$, $R2$, $R4$ and $R8$—sensing members and relay coils of the corresponding binary positions (places)

encoding is the large amount of diodes required. The quantity can be reduced by applying scaling of the measured values. For example, the circuit mentioned above can be transformed by replacing the linear scale by a circular scale made up of two disks with a transmission ratio of 1 : 4. This will require only four diodes instead of 28 (see Fig. 168).

Transducers working on the principle of scale conversion are called *multiple-position* (*multiple-place*) transducers.

A characteristic feature of encoding by reading a field of code combinations is the possibility of occurrence of spurious signals at points of the scale where, in changing the digits in the code, combinations may be formed that repeat the combination of some other, but not the adjacent, point. For instance, in going over in the binary code from number 3 to number 4, a spurious signal for number 7 may appear. Spurious signals appear due to the finite

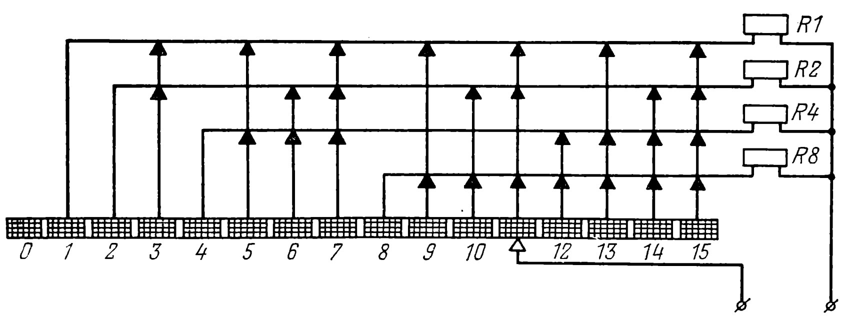


Fig. 167. Encoding by converting the scale quanta into binary code combinations by means of a diode connection matrix

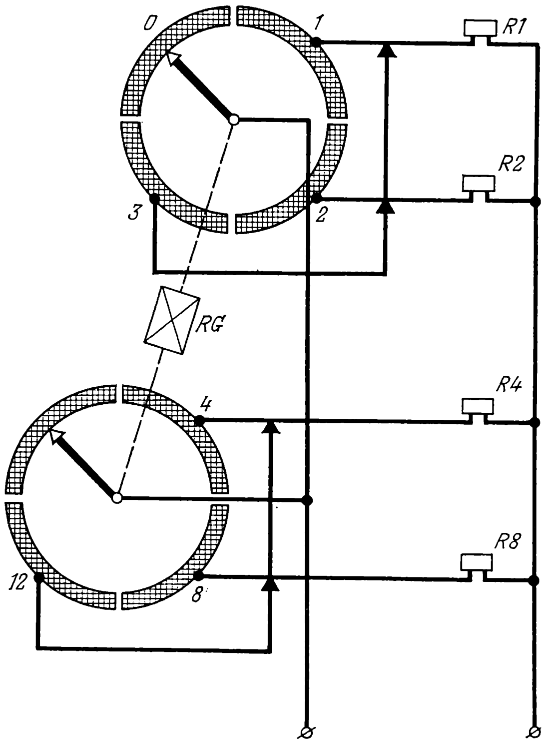


Fig. 168. Scale conversion method:
RG—reducing gear with a 1 : 4 transmission ratio

dimensions of the measuring scales and/or the reading members (for example, brushes).

Various methods are used for eliminating spurious signals:

1. *Interruption method.* This method eliminates undesirable sections of the code combination field (scales) by interrupting the transmission of signals from the transducer in passing from one digit of the code to the next, for example by incorporating a fixing sprocket (Fig. 169) in the design. Reading takes place only when the locking member enters a tooth space of the sprocket.

2. *Method of logical selection of sensing members.* Used here are contacting (brushes) or noncontacting sensing members having two zones, shifted in the form of the letter V, for reading the code combination field. Figure 170 illustrates a circuit for logical selection of brushes, sometimes called the V-reading circuit for the binary code. The connection of brushes Sm_1 or Sm'_1

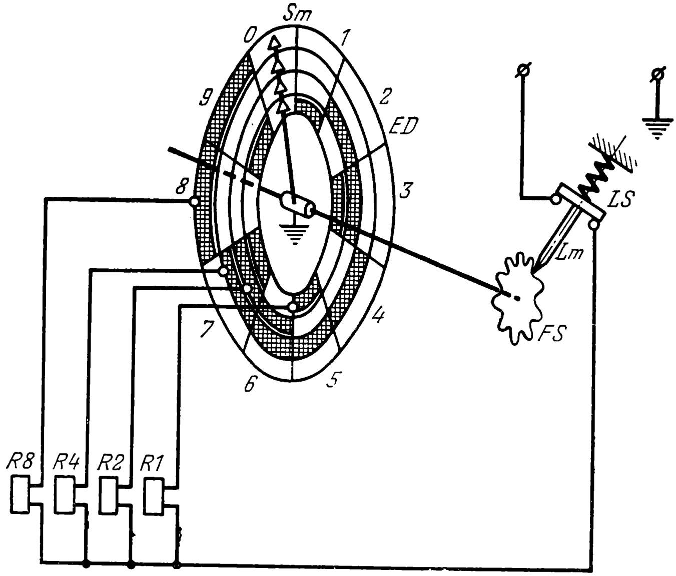


Fig. 169. Diagram of a transducer with a fixing sprocket used for eliminating spurious signals by discretization:

ED—encoded disk; *Sm*—brushes; *FS*—fixing sprocket; *Lm*—locking member linked to the limit switch *LS*

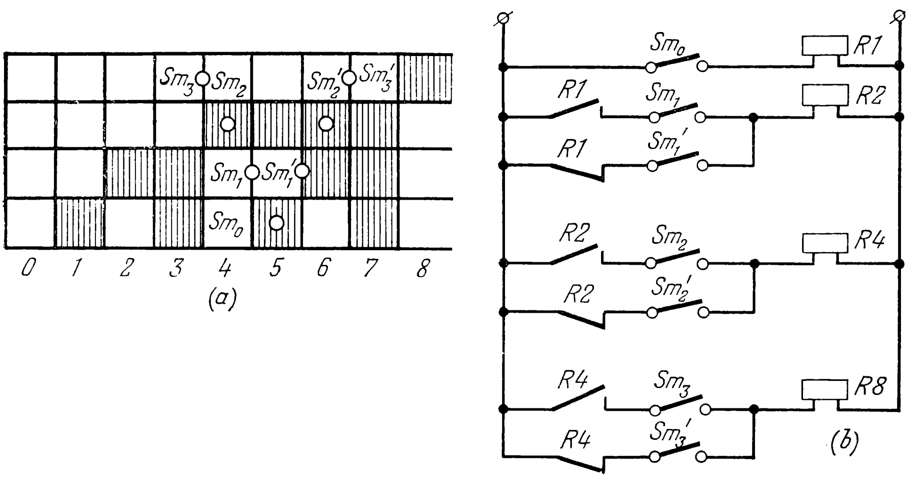


Fig. 170. Circuit for eliminating spurious signals by logical selection of sensing members:

(*a*) field of code combinations with sensing members arranged in the form of the letter V; (*b*) control circuit

is determined by the scale of the lower position (place). If relay *R1* is closed (its coil being energized through brush Sm_0) brush Sm_2 is brought into operation; relay *R2* determines the selection of brushes Sm_2 or Sm_2', etc.

3. *Method of employing special code systems.* The code combinations can be distributed so that in conversion from any digit on the scale, the change in signals from the code combinations will take place in one position (place) only. The conversion error in this case will not exceed one elementary quantum. This property is possessed by *cyclic code systems*, for instance, the Gray code (Fig. 150).

In designing transducers they should always be checked for the possibility of the occurrence of spurious signals.

11-3. Principle of Analog Feedback Transducers

Voltage, frequency or phase levels can be used as electric analogs of the position or displacement of the operative units. Here, the spatial quantities, which are continuous, are checked by the analog transducers without converting them into numerical values. To ensure the proper functioning of the circuit, it is necessary, however, to co-ordinate the numerical programme, i.e., digital information, with the analog information received from the feedback transducers. This can be accomplished either in the control units of the machine tool or separately from the machine in preparing the programme.

In accordance with the chosen analogs of the spatial quantities, potentiometers, capacity or phase pickups (rotary resolvers, selsyns, etc.) can be used as feedback transducers.

Wire potentiometers are contacting transducers and their resolving capacity is limited by the distance between the turns (for a wire size of 0.1 mm the resolution does not exceed 0.1 mm, and they can be called analog transducers with continuous information only arbitrarily). To raise the resolving capacity of transducers, scale conversion can be performed, use being made of multiple-position potentiometers with a speed-up transmission to the lower positions (places). One factor that must be taken into consideration is the poorer contact of the slide with the potentiometer coil at the higher speed of reading in the lower position of the potentiometer (if the speed of the operative unit remains constant). Potentiometer transducers are simple but not very dependable and insufficiently accurate in operation. Therefore they are rarely employed in machine tools. The principle of a phase pickup can be illustrated by considering the following elementary model.

Two coils are arranged side by side (Fig. 171). A sine voltage $U_A = U \sin \omega t$ is applied to coil A. The current appearing in the coil produces a pulsating magnetic flux which induces an e.m.f. in secondary coil B, arranged along the flux. In coil B, $U_B = U_1 \sin \omega t$ (Fig. 171*a*). This device

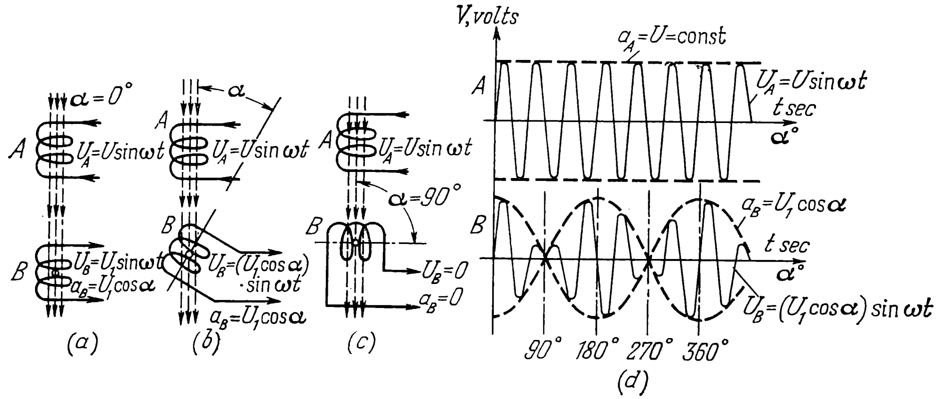


Fig. 171. Inducing an e.m.f. in a coil inclined at different angles in respect to a pulsating magnetic flux:

(a), (b) and (c) various positions of the coil; (d) curves of the applied and output (induced) voltages

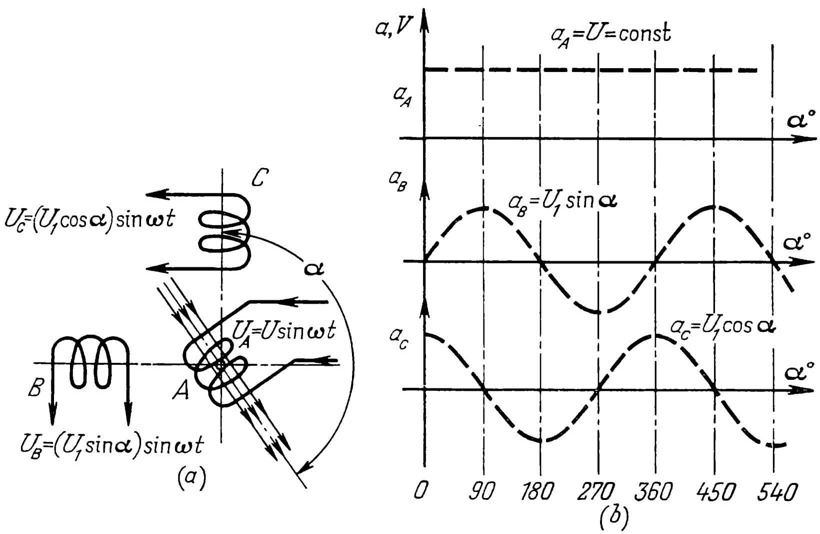


Fig. 172. Inducing an e.m.f. in two coils arranged at 90° by a pulsating magnetic flux inclined at various angles to the coils:

(a) arrangement of the coils; (b) curves of the amplitude values of the induced e.m.f.'s

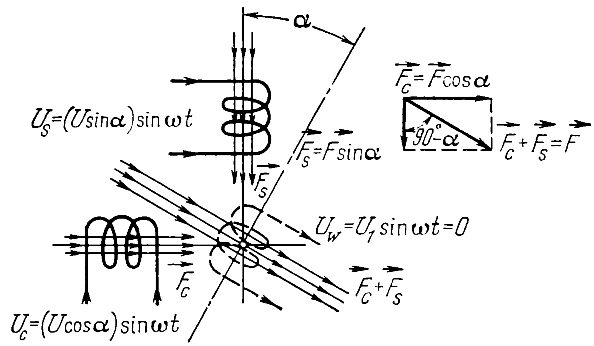


Fig. 173. Principle of the rotary resolver (rotary transformer)

resembles an ordinary transformer with a transformation ratio determined by the ratio of the number of turns of the primary and secondary coils. Next, we begin to turn the secondary coil. When it has been turned through an angle α (Fig. 171*b*) the induced e.m.f. will be smaller and will be determined by the relationship $U_B = (U_1 \cos \alpha) \sin \omega t$. If coil B is turned through an angle $\alpha = 90°$ (Fig. 171*c*), the pulsating magnetic flux ceases to cut the turns of the coil (it begins to merely slide along them) and no e.m.f. will be induced, i. e., $U_B = 0$.

The curves in Fig. 171*d* show the voltage applied to coil A and that taken from coil B at various positions corresponding to $\frac{5}{4}$ of a revolution.

If, now, a sine voltage is applied to a rotating coil, and two other coils arranged at right angles are stationary (Fig. 172*a*), sine voltages can be taken from coils B and C when coil A is rotated. The amplitude values of these voltages will vary with the angle of rotation α. Upon uniform rotation of coil A, the e.m.f. induced in coils B and C will vary according to sine and cosine laws, respectively (Fig. 172*b*).

Next, the coils are connected in another manner so that an alternating voltage with amplitude values varying according to sine and cosine laws is applied to the stationary coils (Fig. 173). In this case, a rotating and pulsating magnetic field is produced which induces e.m.f. in the movable coil. The amplitude value of the e.m.f. at each moment of time depends upon the relation of the phases of the magnetic field and the position of the movable coil (angle α). A variation in this relation produces a phase-modulated signal in the form of a phase-lead or phase-lag voltage, proportional to the deviation or mismatch. Such a device is called a *rotary resolver* (or *rotary transformer*) and can be employed as a transducer in control systems with *phase modulation*. A similar device, having the three coils arranged at 120° to one another in the stator, is called a *selsyn*.

CHAPTER 12

STRUCTURE OF CONTROL CIRCUITS IN FINITE POSITIONING SYSTEMS

Depending upon the method used for encoding the scales of the feedback transducers (counting or reading principle), two types of control circuits are employed in closed-loop systems: counting and coincidence circuits.

Counting circuits are used for the storage of data on the number of pulses contained in the programme and for counting the pulses transmitted by the transducer. They may be made up of selectors, relays, counters with auxiliary step motors or contactless elements, i.e., electronic tubes and semiconductor or magnetic elements. The structure of the circuit is determined by the chosen system of notation for input of the programme and for verifying the position of the operative units.

We may consider, as an example, a relay counting system with binary coding of numerical information.

The circuit consists of a series of binary counters or cells (Fig. 174). One such cell corresponds to one binary position, or place. The open state of the relay corresponds to 0 and the closed state to 1 in the given binary place. In the first cell, shown in the diagram in the 0 state, capacitor C is connected to the source of supply $+E_2$ through normally closed contacts R_1 and P, and is positively charged. Suppose that upon travel of the operative unit the first pulse is transmitted by the feedback transducer and contacts P are reversed. Capacitor C is now connected to the coil of relay R_1. Relay R_1 is closed, its coil being energized by energy stored up by the capacitor, a holding circuit being set up from power supply $+E_1$ through the normally open contact of the relay.

The second normally open contact of the relay prepares for connecting the capacitor to the power supply $-E_3$. When relay R_1 is closed, the cell is in state 1. Simultaneously, the third normally open contact of R_1 connects the capacitor in the second cell to power supply $+E_2$. Upon further travel of the operative unit, the action of the first pulse ceases and contacts P of the transducer return to the state shown in the diagram. The capacitor of the first cell will be negatively charged through the normally closed contact P. Upon the transmission of the second pulse by the feedback transducer, the normally open contact P is closed again and connects the capacitor to the coil of relay R_1. Now, however, the capacitor is negatively charged

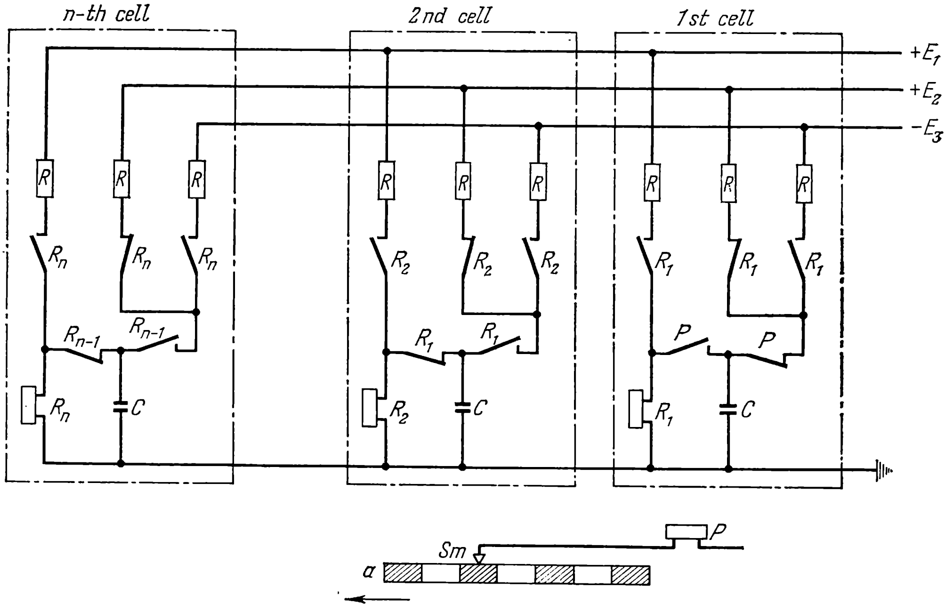


Fig. 174. Circuit diagram of a binary relay pulse counter:
a—scale of the feedback transducer, consisting of current-conducting and insulated sections; *Sm* — sensing member (brush) reading the second pulse

and the current passing through the relay coil opposes the holding current. When these two currents become equal and their sum becomes 0, relay R_1 opens, the first cell goes over to the 0 state and transmits a pulse to the adjacent, higher-order cell since the normally closed contact R_1, located in the second cell, closes relay R_2 with the aid of the previously positively charged capacitor. The second cell goes over to the 1 state.

After this the process is repeated: upon receiving the third pulse, relays R_1 and R_2 are closed, while the fourth pulse opens relay R_1 which, in turn, with its second normally closed contact, connects the coil of relay R_2 to the negatively charged capacitor to break its holding circuit. At this a pulse will be transmitted to the adjacent third cell of the higher-order place. The state of the relays will now correspond to the binary number 100 which means the number 4 in the decimal system of notation.

The purpose of resistor R is to limit the short-circuit current in going over from 1 to 0, in charging the capacitor and also in possible overlapping of the contacts.

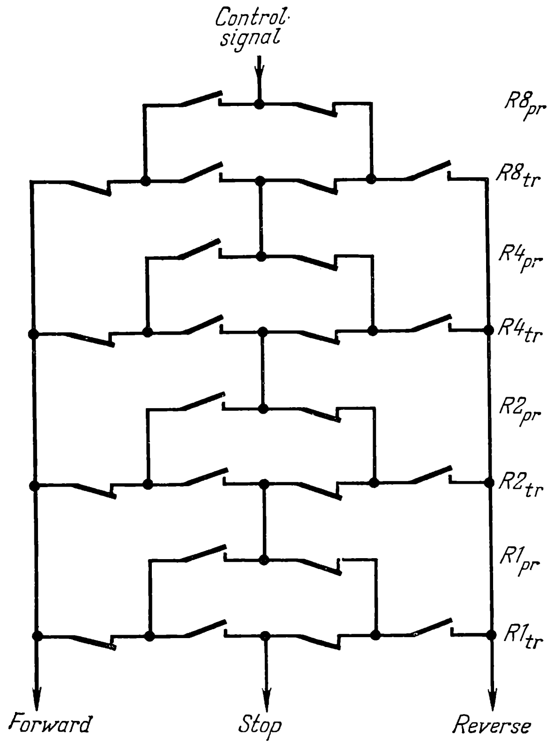


Fig. 175. Circuit for automatically determining (from numerical information in the binary system of notation) the direction of operative unit travel (the coils of the corresponding relays are not shown)

Relay counters can operate at a pulse frequency of 15 to 20 per sec, and up to 50 per sec if polarized relays are employed. Electronic binary counters, consisting of triggers, i.e., contactless elements with two stable states (circuits with electronic tubes or transistors), can operate with a count frequency of several ten thousands hertz (cps). Besides binary counters, decimal counters, made up of decatrons or other contactless elements, can be employed.

Coincidence circuits are used as systems in which the encoding of scales is based on the reading principle. One such circuit was shown in Fig. 147. The accuracy of such a circuit depends upon the accuracy with which the operative unit is stopped by a command from the measuring scale of the lower-order place.

The direction of travel of the operative units is usually given in systems with coincidence circuits since the coincidence circuit itself does not de-

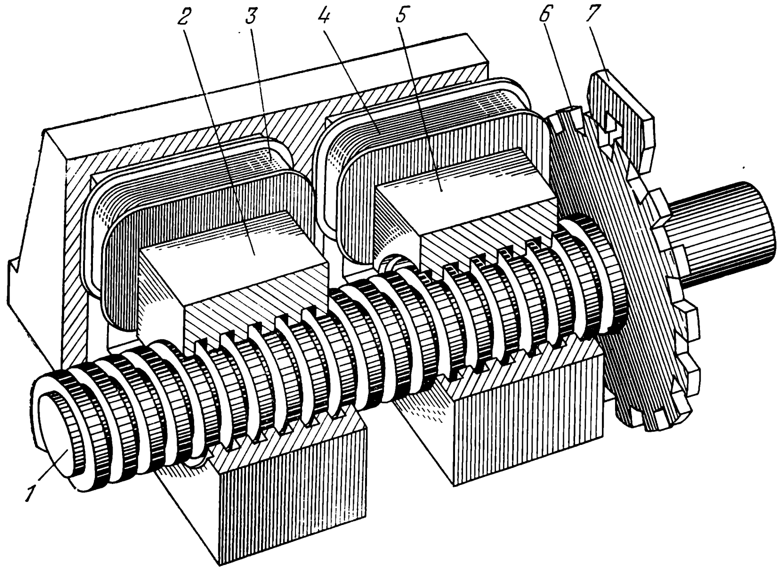


Fig. 176. Inductive transducer (differential transformer):

1—measuring screw; 2 and 5—nuts shifted by one-fourth pitch in reference to each other; 3 and 4—coils of the measuring device; 6 and 7—toothed disk and shoe of the supplementary measuring device for reading fractions of the pitch

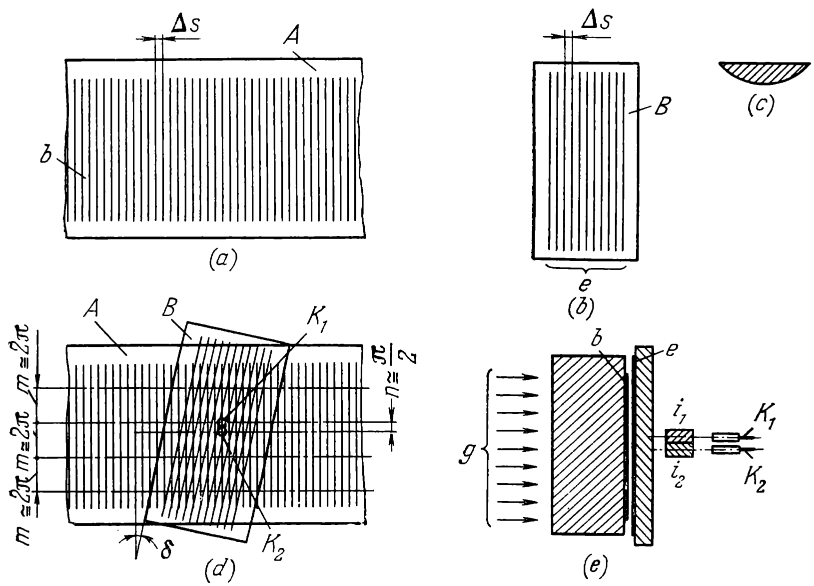


Fig. 177. Principle of the Ferranti photoelectric transducer with diffraction gratings for forming a Moiré fringe pattern:

(*a*) main transparent scale *A* with lines *b* spaced at ΔS; (*b*) short auxiliary transparent scale *B* with lines *e* spaced at ΔS; (*c*) cylindrical collimating lens (i_1, i_2); (*d*) front view and (*e*) side view of the transducer; δ—angle of inclination of the auxiliary scale; $m \propto \Delta S$—distance between the bands of the Moiré fringe patterns; *n*—distance between photodiodes K_1 and K_2; *g*—parallel light beam

termine in which direction the specified displacement is to be executed. However, in some cases, this command lends itself to automation. Thus, for instance, the circuit illustrated in Fig. 175 can be applied under the condition that information is entered and the encoded scale of the transducer is designed in the binary system of notation. Signals of the programme close relay R_{pr} while those of the transducer close relay R_{tr}. Depending on which number is larger, the signal passes along control channel FORWARD or REVERSE. When the number of signals coincide, a STOP command is transmitted (the circuit is shown in this state).

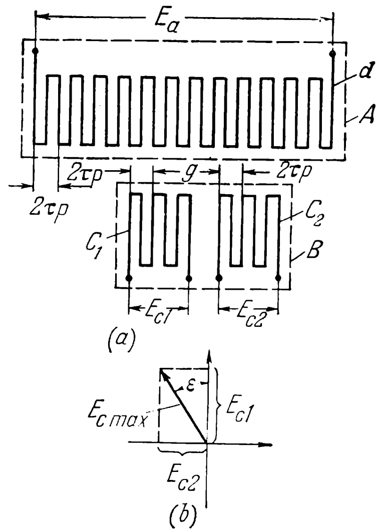


Fig. 178. Principle of the linear Inductosyn:

(*a*) main reference scale *A* and auxiliary moving scale *B* with current-conducting windings *d*, c_1 and c_2, made by the printed-circuit method; E_a—output voltage whose value depends upon the longitudinal position of the auxiliary scale in relation to the reference scale (one scale is in front of the other); τ_p—pole pitch; $g=2n\tau_p \pm \frac{\tau_p}{2}$; where n is a whole number; (*b*) vector diagram where E_{c1} and E_{c2} are the applied voltages (usually with a frequency from 1 to 20 kcps) with a phase shift of 90°; $E_{c\,max}$—total voltage induced by the two windings of the auxiliary scale

Systems of executing the programme can be *single-counting*, with one feedback transducer for each co-ordinate, or *double-counting*. In double-counting systems there are two control systems for each co-ordinate: one for coarse measurement of the displacement and the other for fine measurement. The corresponding transducers have scales with different values of the elementary quantum. In this case, in transducers operating according to a coincidence circuit, the full working range of the fine-measuring transducer is made equal to a single elementary quantum of the coarse-measuring transducer. This enables a high resolution to be obtained with transducers of relatively simple construction (the construction is simplified due to the more favourable relationship between the measuring range and the magnitude of the elementary quantum, i.e., feedback resolution).

In counting control systems (see the description of the N/C lathe developed by the Lenpoligraphmash Plant. p. 248) double-counting systems are employed to reduce the counting rate and the amount of counting cells required.

In designing the numerical controls of a high-accuracy machine tool with a precision system for measuring displacement, it proves expedient to combine a double-counting N/C system with the main measuring system of the machine. As a rule, in such arrangements the system of programming for

course setting provides for traversing the operative unit of the machine to a position within one division of the precision scale of the main measuring system, while the system of programming the fine positioning prepares for precisely stopping the operative unit in accordance with the precision scale (for more details see the description of the N/C jig borer developed by the Moscow Jig-Boring Machine Plant, p. 253).

Used as precision scales in these machines are metal or glass scales equipped with systems of optical magnification having special photocells for making the readings, as in the jig borers of the Moscow and Sverdlov plants. Measuring screws with inductive sensitive heads, developed by the Kirov Plant of Odessa (Fig. 176), are also used as scales in the USSR. Scales with diffraction gratings, such as the Ferranti system in Great Britain (Fig. 177) and the so-called Inductosyn of the Farrand Controls Inc. of the USA (Fig. 178) are finding application in other countries.

CHAPTER 13

EXAMPLES OF MACHINE TOOLS WITH FINITE POSITIONING N/C SYSTEMS

The previously mentioned **N/C semiautomatic lathes, models 1712П and 1722П,** have control systems operating on the principle of a coincidence circuit (see Sec. 9-7 and Fig. 179). These lathes are equipped with circular contacting three-place decimal feedback transducers (Fig. 180) designed for scale conversion. The scales for units and tens of millimetres are on one disk; the hundreds scale is on the second disk. The gearing between the contact disk for units and tens and the disk for hundreds is of the intermittent type with a ratio of 1 : 10. Hence, spurious signals, occurring sometimes when the figures change simultaneously in two places (units and hundreds), are eliminated by the interruption method (see p. 236). Gear *2* has 20 teeth while gear *1* (having the same pitch diameter as gear *2*) has only one tooth space formed by the sides of two adjacent teeth; all the other teeth have been cut away. Gear *3*, with 6 teeth on its right end, meshes with gear *1* with its left end on which only 3 teeth have been left, the others (every other tooth) having been cut away. Thus, the ratio between the contact disks is $\frac{1}{3} \times \frac{6}{20} = \frac{1}{10}$. The contacts for the units and tens are made in the form of ten segments *4* for the tens, and a system of contacts *5* and brushes *6*, designed on the vernier principle, for the units. Instead of one hundred contacts for the units which would have to be arranged on the units-tens disk (for ten tens, one hundred units would need to be counted off), only ten contacts are arranged on the disk (with intervals of 0.11 of the circumference between the contacts from 0 to 9 and 0.01 of the circumference between contacts 9 and 0). The brush counting off the unit contacts is made with ten contacts, equally spaced at intervals of 0.1 of the circumference, and connected together electrically. Upon a relative rotation of the brush contacts through 0.01 of the circumference, commutation of the unit circuits takes place. Thus, the ten contacts for units are read tenfold during a full revolution.

The permissible current density on the contacts does not exceed 0.1 A per sq mm and the minimum voltage on the contacts is 0.5 V.

The following specifications characterize the numerical controls of the models 1712П and 1722П semiautomatic lathes: difference in dimensions that

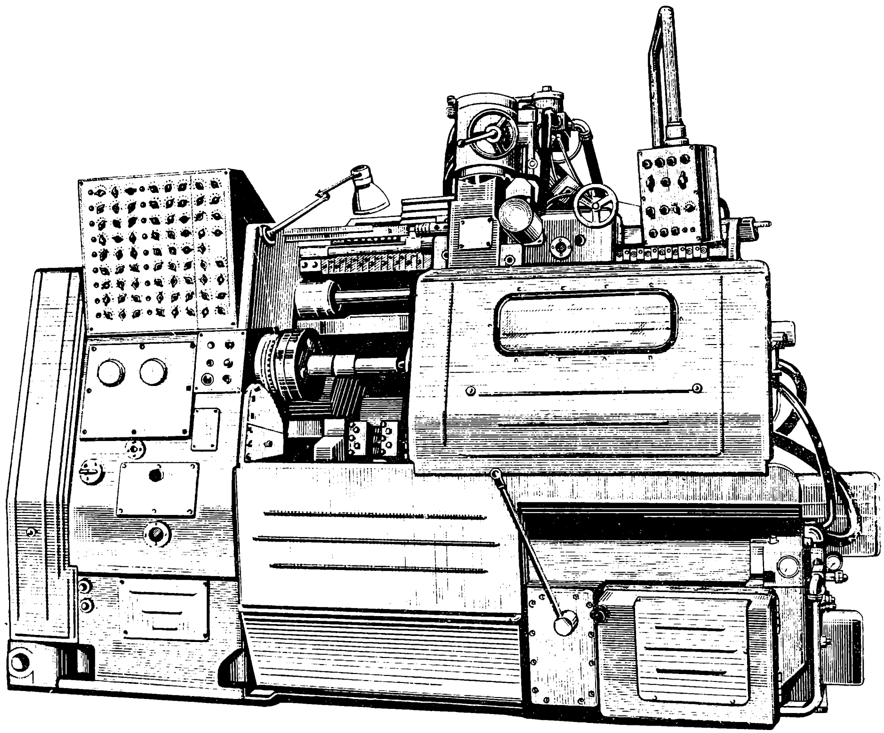


Fig. 179. N/C semiautomatic lathe, model 1712П

can be programmed (resolving capacity)—1 mm in length and 0.2 mm in diameter; accuracy of automatic machining—0.5 mm in length and 0.1 mm in diameter. Stepped shafts having up to nine different diameters and nine different step lengths can be machined. The machine can be set up with a programme in several minutes. In the model 1712П lathe, the maximum longitudinal travel of the carriage is 500 mm; the maximum cross travel of the slide is 75 mm.

N/C lathe, model ТП-1М, developed by the Lenpoligraphmash Plant. This lathe is intended for turning cylindrical stepped shafts with two tools clamped in the front and rear toolholders.

The longitudinal carriage is powered from the feed gearbox through electromagnetic clutch *1* (Fig. 181) and the feed rod. Rapid traverse of the carriage is effected by a separate electric motor *8* mounted at the right end of

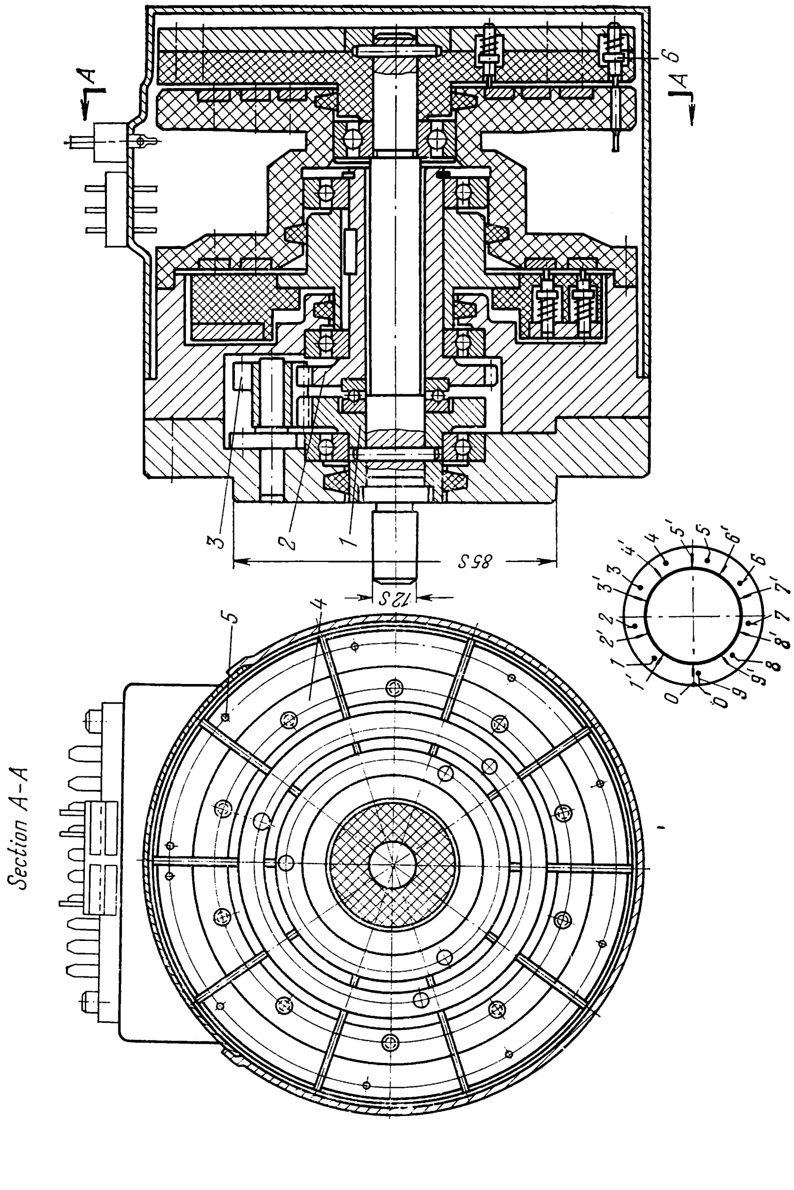


Fig. 180. Contacting feedback transducer of semiautomatic lathes, models 1742П and 1722П: assembly drawing and diagram of the vernier for reading units

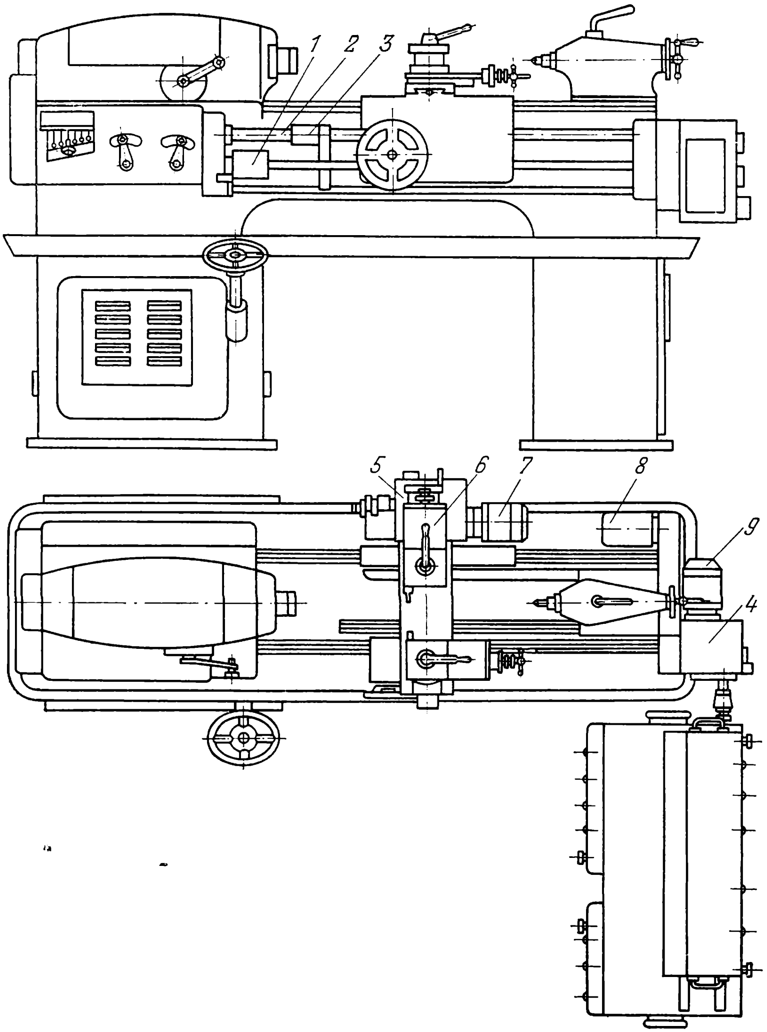


Fig. 181. General view of the model TП-1M N/C lathe

the feed rod. The position of the longitudinal carriage is controlled by a simple electric copying device consisting of movable stop *3*, mounted on lead screw *2*, and a limit switch secured to the left end face of the apron

(not shown in Fig. 181). When the pin of the limit switch is pressed, clutch *1* is disengaged, after a certain time delay, and the carriage runs up to a positive stop by inertia and stops. The use of a positive stop for controlling carriage travel increases the machining accuracy.

The movable stop is positioned by the lead screw which, in this lathe, is connected neither to the feed gearbox nor the apron of the carriage, but is powered from a special drive *4* by electric motor *9*. The screw of cross slide *6* is not connected to the apron either, but has an independent drive *5* powered by electric motor *7*. The drives of the screws have almost identical kinematic trains (Fig. 182). From the electric motor, rotation is transmitted through gears *7*, *6*, *5* and *4* to gears *3*, *2* and *1*. Rotation can be transmitted further to three intermediate shafts and, by means of spiral gears *8*, *9*, *10* and *11*, worm gearing *K2-12* and two bevel-gear differentials *DF-1* and *DF-2*, summing up the motion, to the screw. The shafts are engaged by means of single-revolution clutches *C1*, *C2* and *C3* actuated by solenoids *Sd1*, *Sd2* and *Sd3*. Each shaft has a feedback transducer (*FT-1*, *FT-2* and *FT-3*) which is a single-contact pulse transmitter, operating by the counting principle. One revolution of shaft *I* corresponds to a movable stop displacement of +0.1 mm; one revolution of shaft *II* to a displacement of +1 mm and of shaft *III* to —1 mm (the signs indicate the direction of travel of the unit). The corresponding displacement values for the cross slide are +0.01 mm, +0.2 mm and —0.2 mm.

In order to programme the displacement of the movable stop (i.e., of the longitudinal carriage) or the cross slide, a corresponding number of revolutions is to be imparted to shafts *I* and *II* or shaft *III*. This information is recorded in a binary code on a punched card. When the card is read, after momentary closure of contact *S* (Fig. 183), the information is fed into the binary relay counter which operates by countdown. The principle of the counter was described above (p. 241). The circuit will be in the following states: initial state, R_8—open (0); R_4—closed (1); R_2—open (0) and R_1—closed (1), corresponding to the number 5 (0101); from the first pulse 0100⟶4; from the second 0011⟶3; from the third 0010⟶2; from the fourth 0001⟶1, and from the fifth 0000⟶0.

After each shaft has made the specified number of revolutions, and signals from the feedback transducers are transmitted to the counter, all the cells (places) of the counter go over to the 0 state and their relays open the normally open contacts in the circuit of coil R_c. This disengages the single-revolution clutch of the corresponding shaft. The diagram shows the circuit for the transducer and the single-revolution clutch for shaft *I*.

In the model TП-1M lathe, the complete data memory and comparison unit for longitudinal travel of the carriage consists of two relay counters: one 8-place counter for fixing "large" pulses with a division value of 1 mm and enabling up to 255 pulses to be fixed, i.e., 255 mm, and one 4-place

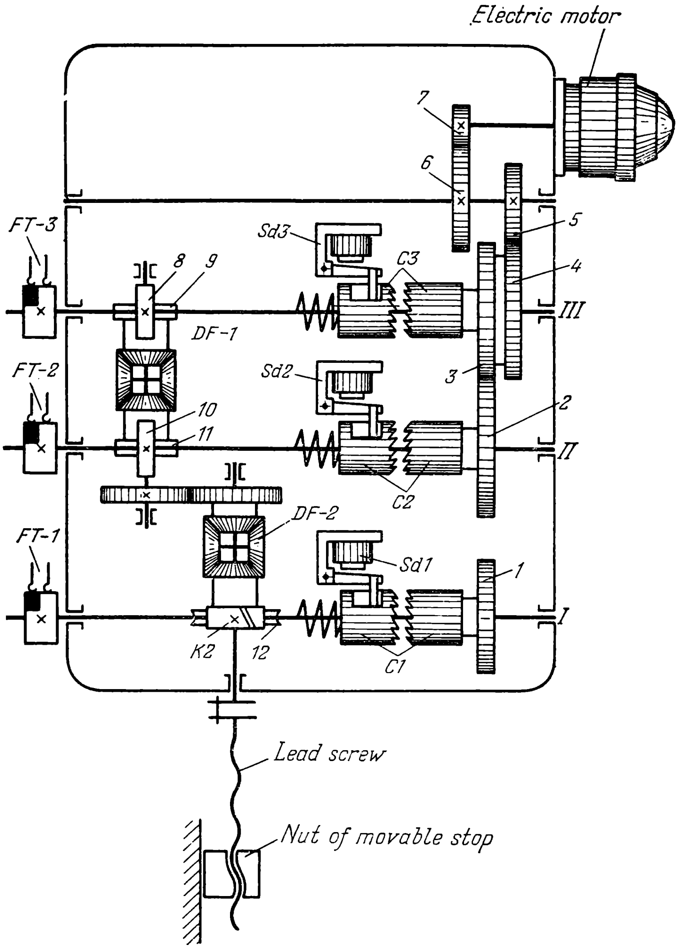


Fig. 182. Kinematic diagram of the screw drive mechanism of the model ТП-1М lathe

counter for "small" pulses with a division value of 0.1 mm. Each subsequent displacement of the operative unit is counted from the preceding displacement, i.e., from a new zero reading.

As is evident from the description, the programming system of this lathe is of the double-counting type based on the counting principle. The resolution of the lathe in operation with numerical controls is 0.1 mm along the

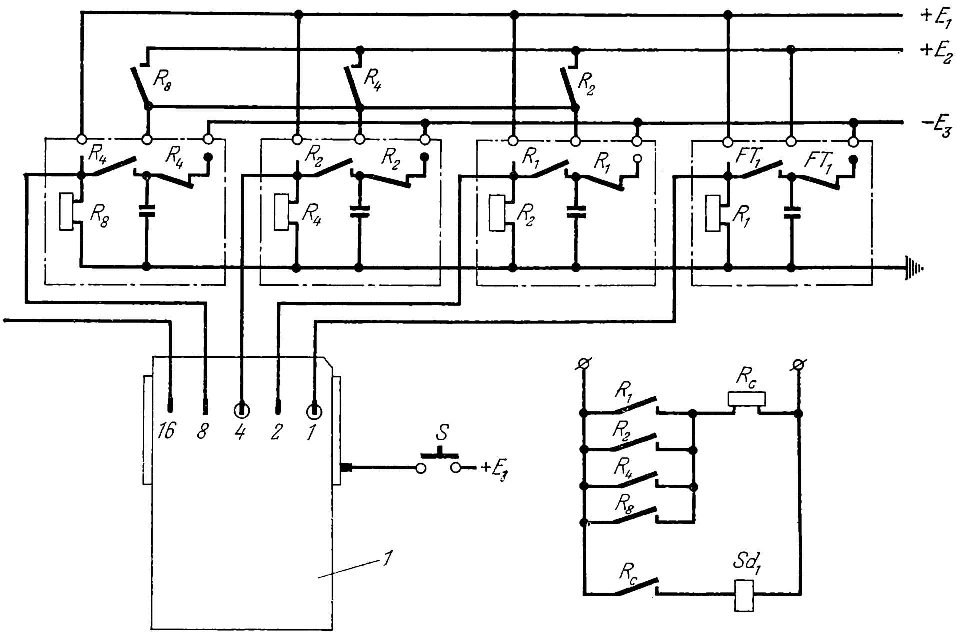


Fig. 183. Circuit diagram of the binary relay counter operating by countdown in the model ТП-1М lathe (the circuit is shown in the initial position before data is read): 1—punched card with the recorded number 5; $Sd1$—coil of the solenoid for operating the single-revolution clutch

length and 0.02 mm on the diameter of the workpiece. The machining accuracy is 0.06 mm for length dimensions and 0.03 mm for diameters. The height of centres is 150 mm and the distance between centres is 750 mm.

Jig-Boring Machines of the Moscow Jig-Boring Machine Plant. The model 2А450П N/C jig borer is designed for operation from punched cards containing data on the co-ordinate settings of the workpiece for boring holes dimensioned in the rectangular co-ordinate system. The first experimental model was designed on the basis of the general-purpose jig borer, model 2A450 (described in Sec. 22-3, Vol. 1), which is lot-produced in this plant. The control system for each co-ordinate is of the double-counting closed-loop type with feedback transducers designed on the principle of reading a field of code combinations. Each of the four control blocks (two for each co-ordinate axis) is based on the same coincidence circuit and is equipped with a unified three-place feedback transducer (with scale conversion).

The transducers are linear-displacement-to-digital converters, the numerical code being recorded in the form of definitely arranged holes in thin aluminium disks. They have encoded disks for units, tens and hundreds, a decimally-coded system being employed. The "two-out-of-five" code is used for units and tens. To avoid the occurrence of spurious signals, the units scale is quantized by specifying discrete points. For this purpose the openings in the disk for the two-out-of-five code are narrow and the commands are taken off during a tenth of their value. A cyclic "two-out-of-seven" code is used for the hundreds disk (likewise to avoid spurious signals). The transmission between the positions (places) is continuous with the ratio of 10 : 1 between the tens and units disks and approximately 1 : 10 between the tens and hundreds disks. The transducers are of the contactless type and employ photodiodes as the sensing ("reading") members. The construction of the transducer is illustrated in Fig. 184.

The jig borer has two operative units for co-ordinate positioning according to the programme, namely: the table (co-ordinate x) and the saddle (co-ordinate y), and two auxiliary travelling units which are also programmed as to position. These last two are the slide with the photoelectric transducer of the table position screen and the slide with the photoelectric transducer of the saddle position screen on which the fractional part of the reading (fractions of a millimetre in microns) is shown for the co-ordinate being set. The final positioning of the operative units is done to precision scales with 1-mm divisions on which the main reading and measuring system of these jig borers is based. The principle of setting co-ordinates along the x-axis (table) with programmed controls is shown in Fig. 185. The exact dimension is read by means of an optical system which projects an image of the scale graduations on the screen (optical magnification 125X). The scale graduations are sensed by the photoelectric transducers of the table and saddle position screens. The auxiliary units (slides with the photoelectric transducers) consist of drive servomotors, worm gearing and screw and nut pairs which traverse the slides with the photoelectric transducers along the screens. This device serves as a vernier for reading the fractional part of the dimension.

An eighty-column punched card is used to record the positioning programme (data are recorded along the rows). One row specifies the positions of the operative units (table and saddle) in whole millimetres of the co-ordinate dimensions (positioning accuracy of approximately 0.3 mm), while the second row specifies the positions of the auxiliary units (slides of the table and saddle screens), corresponding to the fractional part of the co-ordinate dimensions, i.e., microns. Since the reading of the data on the whole and fractional millimetres of the co-ordinate dimension need not be simultaneous, separate memory units are not required. The feedback transducers, checking the whole millimetres of the co-ordinate dimensions, are linked to the operative units by rack-and-pinion drives in which the racks are sections of

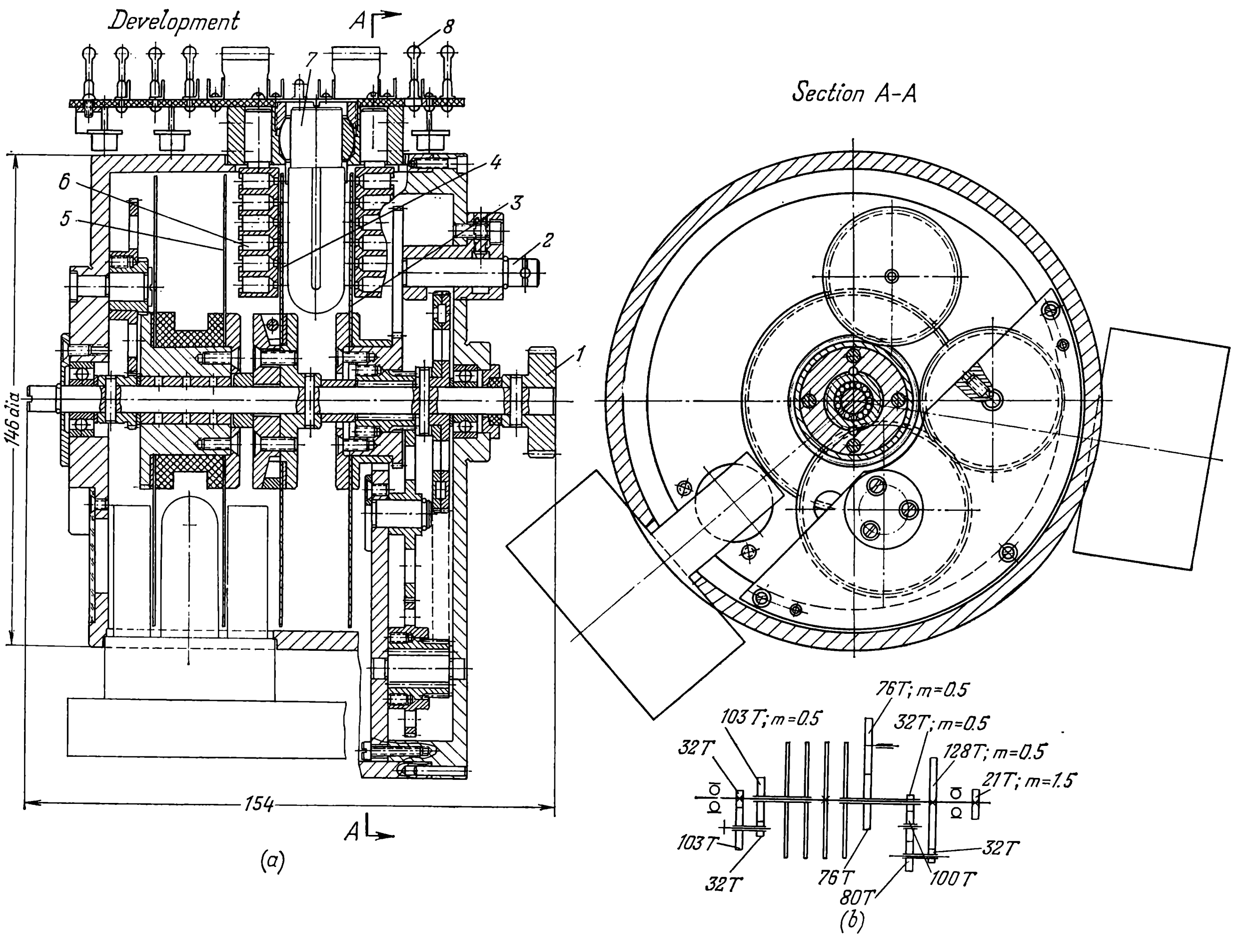


Fig. 184. Contactless feedback transducer with encoded disks in the model 2A450П jig borer:
(*a*) assembly drawing; (*b*) kinematic diagram; *1*—rack pinion (input for table and saddle); *2*—shaft of gear *76T* (input for slides of screens); *3*—encoded units disk; *4*—tens disk; *5*—hundreds disk; *6*—photodiodes, type ФД-3; *7*—illuminating lamp, 7 W; *8*—photodiode amplifier

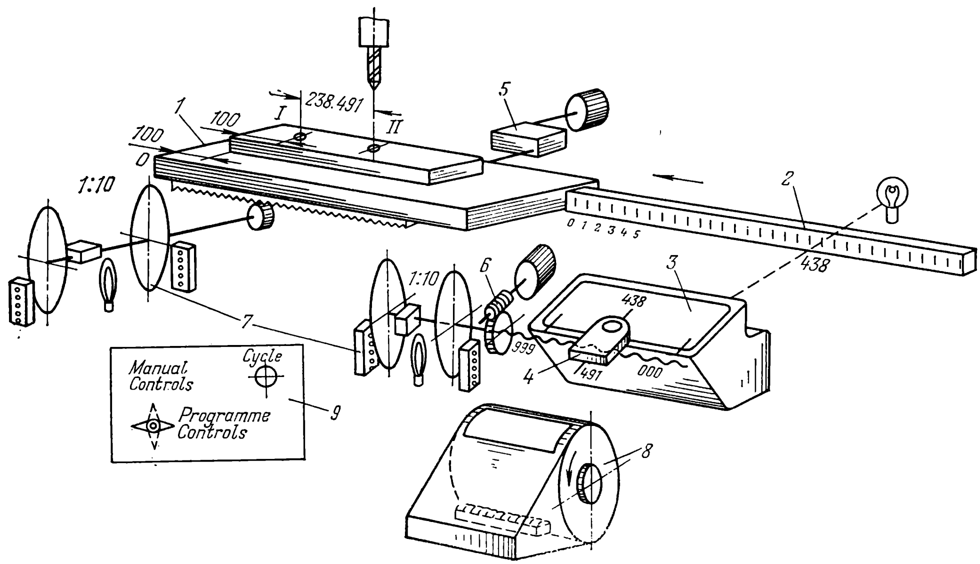


Fig. 185. Diagram showing co-ordinate positioning of the table by programme controls in the model 2A450П jig borer:

1—table with workpiece; *2*—precision scale of the main measuring system; *3*—screen of the optical reading system; *4*—slide of screen with photoelectric transducer; *5*—table drive; *6*—drive of screen slide; *7*—feedback transducers; *8*—device for reading punched cards; *9*—control panel

precisely ground screws. The transducers are mounted in such a way that the housing can be swivelled through a small angle, sufficient for shifting the co-ordinate readings by ± 0.7 mm. This facilitates setting up and enables boring to be performed from locating surfaces with accurate dimensions according to the programme (finished holes may also be used for location). The feedback transducers that check the system of the intradivision reading (the fractional part of the co-ordinate dimension) fix the position of the screen slides. In construction all the transducers are identical but have different inputs. Motion can be transmitted either to the tens disk, for the table and saddle, or to the units disk, for the slides of the screens (see Fig. 184).

The photoelectric transducers of the screens, intended for precisely sensing the scale graduations, are photoresistors secured on special electromechanical modulators (Fig. 186) which are mounted on the slides of the reading screens. The use of a modulator in the system enables the position of the scale graduations to be sensed with high dependability and stability, regardless of the thickness of the graduation (line) and the low speed of travel (less than 1 mm per min) with an accuracy higher than 1 micron. This system was developed by the Moscow Jig-Boring Machine Plant in conjunction

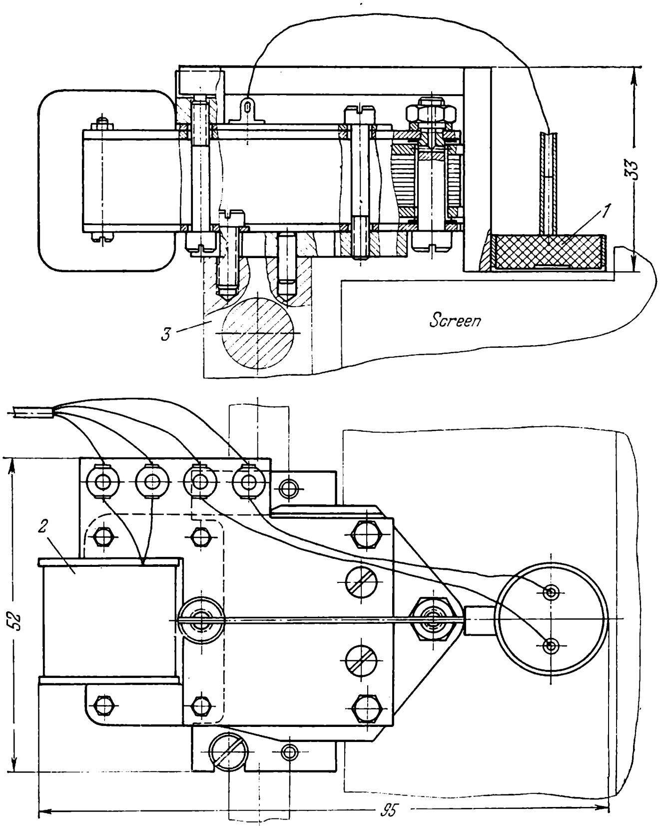


Fig. 186. Photoelectric transducer with a modulator:
1—photoresistor, type ФСД-1; 2—vibrator (electromechanical modulator); 3—slide of the screen

with the Electronics Department of the Moscow Machine Tool Engineering Institute.

To increase the accuracy with which co-ordinate dimensions are set, the final movement of the operative units—the final approach to the point of alignment—is always in the same direction and at a reduced speed (approx. 0.6 mm per min). This is accomplished by the application of a special control circuit with a magnetostrictive drive. The direction in which the values of the figures on the scales increase is taken as the positive direction of travel of the table and saddle.

Since the working speed of travel of the operative units is quite high (1,200 mm per min), and since only one point is programmed each time—

the corresponding whole millimetres part of the co-ordinate dimension, determined by a discrete-action feedback transducer—the travelling units (table and saddle) pass these points by inertia. What might be called the "bracket" method has been used in this machine to bring the operative unit to rest in the required position. This method provides for stepped speed reduction and simultaneous reversal of the table or saddle each time they pass by the programmed point. When the unit reaches the specified position, a command is transmitted for transferring control to the reading of the co-ordinate dimension on the corresponding scale. As a result of several oscillating motions of the table (or saddle) in the vicinity of the programmed point, with a gradual reduction of the speed and a corresponding reduction of overtravel, it is possible, with the aid of the feedback transducer and braking facilities, to fix the programmed position for the whole number of millimetres of the co-ordinate dimension. After this, to achieve alignment with the fractional part of the co-ordinate dimension, a command is transmitted to the operative unit for travel until the scale graduation coincides with the photoelectric transducer of the screen (travel is always in one direction and at a speed of 30 mm per min).

The operative units are automatically switched over to a speed less than 1 mm per min by an arrangement in which the precision scales of N/C jig borers have auxiliary graduations located 40 microns in front of the main graduations which are spaced at 1-mm intervals and serve for precise readings of the co-ordinate dimensions. First the auxiliary graduation enters the zone of action of the photoelectric transducer. At this, the table (or saddle) drive motor is switched off and the drive from the magnetostrictive actuator is engaged to traverse the table (or saddle) until the main graduation is in alignment with the photoelectric transducer. The travelling unit is now in the precise position and is automatically clamped. The error in co-ordinate positioning to the programme does not exceed 6 or 8 microns.

The preparation of the programme and the machining of workpieces in the model 2A450П jig borer consist of the following stages:

1. In accordance with the part drawing, in which the dimensions have been specified in the rectangular system of co-ordinates, the process-planner-and-programmer assigns the sequence of operations, the co-ordinates of the locating datum surfaces (thereby determining the position of the workpiece on the table), and the co-ordinate points required for machining. All the co-ordinates are given from the zero point of the measuring system of the machine in the form of six-digit numbers.

2. A table of initial data is compiled. It contains the name and number of the part to be machined; the location datum surfaces selected for the machining operations and their co-ordinate dimensions in reference to the zero points of the scales for the table (X-axis) and saddle (Y-axis); the numbers of the operation elements and co-ordinate dimensions of the hole centres,

specifying the direction of travel of the operative unit for each hole (if travel takes place with a reduction in the numerical value of the preceding co-ordinate, the letter *R*—Reverse—is written). The numerical values of the co-ordinates, specified from the zero reading of the jig borer scales, are listed in two columns, at the left for the table (X) and at the right for the saddle (Y), and in two lines. The first line is marked with the letters *SC*, indicating that it refers to the slides of the screens with data on the fractional part of the co-ordinate dimension. The second line contains data on the whole number of millimetres in the co-ordinate dimension and is marked by the letters *TS* (table and saddle). If no travel is required for any co-ordinate the figure 0 is written.

3. From this table, an operator punches the cards using a special perforator with a keyboard input. The perforator has an encoder which automatically converts the decimal system of notation into the code employed in the jig borer.

Data on six different positions of the table and saddle can be recorded on a single punched card.

It is advisable to check the programme recorded on the punched card by repeating the recording of the same programme on a second card and placing the two cards one on the other to see if the punched holes coincide (it is better to have a different person punch the duplicate card). It is good practice, after thus checking the cards, to keep one set as a master.

4. Other information, to be written on the cards, includes the name and number of the part, datum with dimensions, the number of the card and the total number of cards required to machine the workpiece.

5. The finished punched cards with a drawing of the part, process sheet, cutting and measuring tools, blanks and the necessary tooling for setting them up from the datum surfaces are issued to the jig-boring machine operator.

6. In accordance with the directions written on the punched cards concerning the location datum, the operator sets up the tooling and workpiece on the jig borer table.

7. The punched cards are put into the reader in the same sequence that the workpiece is to be machined. Up to eight cards can be put in at one time. This means that the reading mechanism (see Fig. 157) can hold data for 48 different positions of the table and saddle.

8. The selector switch on the control panel is turned to the PROGRAMME CONTROL position (see Fig. 185).

9. The first line on the punched card is read. The slides of the table and saddle screens automatically move to positions corresponding to the values of the fractional parts of the required co-ordinate positions of the table and saddle. After this the mechanism for reading the programme indexes one pitch, preparing the next line of the punched card for reading.

10. When the CYCLE push button is pressed, the table and saddle travel to a position corresponding to the point of intersection of the locating datum surfaces of the workpiece (if it is necessary to check and finally position the fixture for locating the workpiece in accordance with the programme), or the table and saddle travel to the first position for machining.

11. The operator clamps the locating fixture and the workpiece or, if this is not required, he machines the workpiece at the first position.

12. As soon as the table and saddle have reached their programmed positions, the card reading mechanism moves the punched card one pitch and the slides of the screens move to new positions corresponding to the fractional parts of the co-ordinate dimensions for the next machining position.

13. The setting is made for each new position by pressing the CYCLE push button after the screen slides have reached positions corresponding to the fractional part of the dimension for the new position and the programme reading mechanism has indexed one pitch.

The final movement of the operative units to the exact dimension is not checked by the feedback transducers. This movement takes place at low speed, first being effected by an electric motor and then actuated for the last 40 microns of travel by the magnetostrictive drive. This last movement continues until signals are received from the photoresistors (mounted together with the modulators on the slides of the screens) in response to the images of the scale graduations projected on them.

Specifications of Jig Borer, Model 2A450П, Positioning System

Table travel, mm	1,000
Saddle travel, mm	630
Maximum traverse speed, mm per min	1,200
Resolving capacity of the N/C system, microns	1
Repeatability in co-ordinate positioning to a programme, microns	3
Time required to prepare the programme for six positions and to record it on a punched card, min	8

The application of numerical controls in a jig borer increases the production capacity by 15 to 20 per cent and reduces operator fatigue at the same time.

Another model of N/C jig borer, developed by the Moscow Jig-Boring Machine Plant, uses selsyns as feedback transducers. The application of an analog circuit has definite advantages in comparison with discrete systems, one being the possibility of developing a more economical drive for the travelling units. The speed of the operative units can be varied proportionally to the amount of error between the transmitting (input) and feedback selsyns. Hence, the operative unit gradually slows to a full stop as it approaches the programmed position. The application of differential selsyns in the system simplifies "displacement of the zero reading". Drawbacks of this system are the considerable time required to prepare the reading of

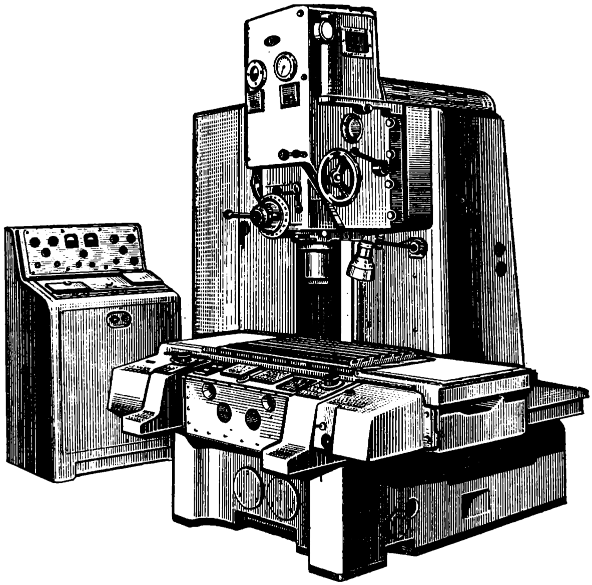


Fig. 187. N/C jig-boring machine, model 2B440П

the fractional part of the co-ordinate dimension and the unwieldy circuit used to convert discrete numerical data, fed in from the programme, into analog data. This is done in the jig borer by means of an auxiliary contact transducer which checks the position of the input selsyn and compares it with the programme, using a coincidence circuit. More advantageous in this respect is a new arrangement, developed by the plant together with the Servo Drive Laboratory of the Institute of Control Engineering. In this circuit, multiple-winding transformers are used as reference input elements, while selsyns are used only as feedback transducers. The connection of the various terminals of the transformers is accomplished directly in accordance with the programme recorded on a punched card by means of the contacts of relays linked to the brushes of the programme reading mechanism.

The N/C jig borer, model 2B440П, operating by a system incorporating multiple-winding transformers, is shown in Fig. 187.

Specifications of the Jig Borer Positioning System

Table travel, mm	710
Saddle travel, mm	400
Maximum traverse speed, mm per min . . .	1,500
Resolving capacity of the N/C system, microns	1
Repeatability in co-ordinate positioning to a programme, microns	2

CHAPTER 14

CONTOURING SYSTEMS OF NUMERICAL CONTROL

14-1. Principle of Operation of Machine Tools with Contouring N/C Systems

To obtain any kind of profile in machining, for example, to turn a taper in a lathe, or to machine a surface that is inclined to the table ways in a milling machine, it is sufficient to traverse the tool in a lathe or the workpiece in a milling machine simultaneously in the longitudinal and cross directions. These motions should take place, not only simultaneously, but properly interrelated over the whole extent of working travel. For instance, to machine a template on the section A-B (Fig. 188) which is a straight line inclined at an angle of 30° to the x-axis, the template must be traversed along this axis approximately twice as fast as along the y-axis, since the corresponding path lengths along the axes are in a ratio of about 1 : 2 ($\tan 30° \cong 0.577$). Evidently, the speed of traverse along the co-ordinate axes should be the same to machine section B-C. Moreover, the rate of feed along the whole profile should not vary greatly if the milled surface is to have correct geometric features and the same surface finish throughout.

As distinguished from finite positioning N/C systems, in contouring systems a simultaneous interrelated process of automatic control along all working co-ordinates proceeds continuously over the whole extent of working travel of the operative units.

It has been mentioned above (Sec. 11-1) that control systems may be of either the closed- or open-loop type. In the first case, provision is made for checking the movements of the operative units by means of feedback transducers, and in the second by means of some special type of drive providing for travel of the operative units in steps whose magnitude is constant. This means that in our example of contour machining, and depending upon the chosen type of control system, the data fed into the machine tool must be converted in such a way as to be continuously compared with data transmitted by the feedback transducer, or these data can be used for continuous control of a step drive. Most convenient for this purpose is the tally system of notation, or unitary code. Here, each "1", i.e., each signal of the tally code corresponds to one elementary displacement of the operative unit.

Let us assume that the distance from A to B (Fig. 188) is equal to 20 mm along the x-axis and to 10 mm along the y-axis. If it is possible to infeed 20 signals, distributed uniformly in time, to the control system for longi-

tudinal travel of the workpiece simultaneously with 10 such signals (1's) for cross travel, and if the drive, continuously controlled by feedback transducers, operates so that one signal will be transmitted by the transducers when the workpiece travels 1 mm along the co-ordinate axes, the specified programme will be realized.

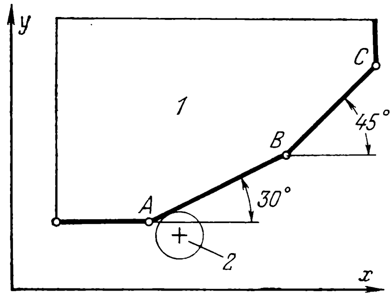


Fig. 188. Contour machining a template in a milling machine:
1—workpiece; *2*—cutter (end mill); *A-B-C*—section milled

The principal scheme is simplified in an open-loop system. In this case, it is sufficient for the drive to make 20 and 10 steps (1 mm per step) simultaneously along the x- and y-axes, respectively, likewise with uniformly distributed signals. The following condition must be complied with (in our example) regardless of the system of N/C contouring applied: for each two signals of the programme for travel along the x-axis, one signal for travel along the y-axis should be received (in actual practice, the section A-B would be machined too roughly, having steps up to 1 mm in size).

A *unitary code* in an N/C system is a sequence of pulses in which each pulse corresponds to a displacement of the operative unit through one step. The magnitude of the displacement of the operative unit from one pulse is called the *pulse value* (PV).

To obtain a higher class of surface finish it is necessary to employ a machine tool with a contouring N/C system having a smaller pulse value.

If in the example mentioned above a pulse value of 0.01 mm is used, the corresponding displacements of the operative units should be programmed as 2,000 and 1,000 hundredths of a millimetre, and the machined surface will comply with high requirements as to finish.

14-2. Preparing a Programme for Contouring N/C Systems

The preparation of a programme in machine tools with contouring controls is a much more complicated process than in those with finite positioning controls. The simplest procedure in milling a curvilinear profile with the aid of numerical data is to reproduce the required path of motion of the centre of the cutter (end mill). This path will be the equidistant curve of the given profile, taking into account the cutter radius. However, if an attempt is made to calculate the co-ordinates of all the points on this curve in sufficient detail, for instance, at intervals of 0.1 mm, not only will an input

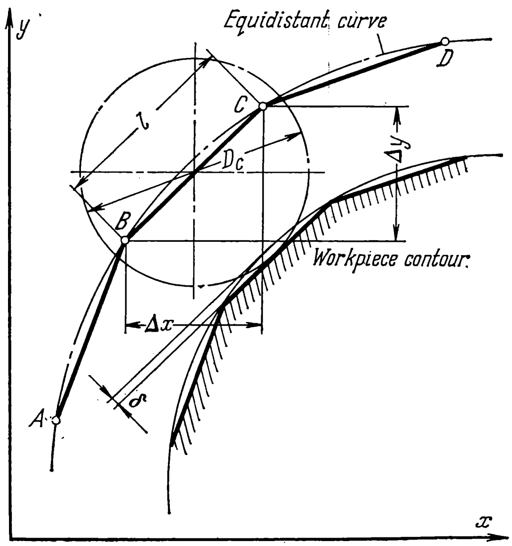


Fig. 189. Piece-wise approximation of a curvilinear contour:

δ—error due to approximation (approximation error), D_c—diameter of the cutter (end mill) used for the given milling operation

medium of very large capacity be needed to record these data, but the computations will require a great deal of time. In actual practice a more efficient procedure is followed.

After determining and drawing the equidistant curves of the profile to be milled, the profile is broken down into a series of segments which are replaced, or approximated, by simpler lines. If a series of straight lines substitute for the curve (Fig. 189), this form of conversion of the contour is called *piece-wise approximation*; if, instead of straight lines, circular arcs are used, the conversion is called *circular-arc approximation*. Error is deliberately introduced when the contour to be machined is approximated. This error is equal to the maximum deviation of the new approximate profile from the given one and is called the *error of approximation* (δ). An approximation should be based on the permissible error of approximation which should be a part of the total permissible machining error for the given machine tool. The tolerance on the approximation error or, as it is called, the *mathematical tolerance*, constitutes practically from 15 to 25 per cent of the total tolerance on the machining errors of the workpiece.

The points common to the new profile obtained by approximation and to the given profile are called *reference points*.

As a result of approximation of the workpiece contour, the path of motion of the cutter centre, the equidistant curve, is altered in a corresponding manner and the reference points are therefore transferred to the equidistant curve (points A, B and C in Fig. 189). Next the increments of the co-ordinates are computed consecutively for each reference point of the equidistant curve. These increments are listed in the calculation sheets directly in the form of a number of pulses. For this purpose, the dimension of each increment is multiplied by the constant multiplier $\frac{1}{h}$ (number of pulses per mm of displacement), where h is the pulse value. For example, if the pulse value is

25 microns, then $\frac{1}{h} = 40$ pulses per mm. To traverse the table 3.75 mm, it is necessary to transmit $3.75 \times 40 = 150$ pulses. The time required to travel over each elementary segment (for instance, from B to C) is determined by the formula $t = \frac{60l}{s}$ or $t = \frac{60\sqrt{\Delta x^2 + \Delta y^2}}{s}$ sec where the increments of the co-ordinates Δx and Δy are expressed in millimetres and the rate of feed along the contour, s, in mm per min. Such information will be quite sufficient to enable special computing devices to convert the numerical encoded programme into a unitary code with any linear or nonlinear relationship of motion of the operative units.

All calculations involved in preparing programmes for machine tools with contouring N/C systems can be expediently done, due to their exceptionally large volume, by electronic digital computers. If such are not available, the calculations can be done manually using only table-type keyboard computers or arithmometers. Punched tape is employed, as a rule, as the primary input medium for machine tools with a contouring N/C system due to the large amount of data required. The data from the calculation sheets are usually recorded on the punched tape in the decimally coded form so that the information for machining each segment between two reference points is given by a command consisting of a definite amount of numerical data characterizing the magnitude, time and direction of displacement along all the co-ordinates participating in the forming of the workpiece. This information makes up the *programme block*.

The mathematical tolerance should not be too small since this will lead to an unjustified increase in the volume of calculations and also increase the number of blocks required. This last, in turn, may increase the probability of failure in operation when the programme is fed into the machine tool.

Since curvilinear profiles of workpieces are frequently made up of segments formed by circular arcs, it may be more efficient to employ circular-arc approximation. Even if nonlinear computing facilities are unavailable, it is possible to compile approximation tables for circles beforehand with the most frequently used radii of machining and to apply these finished data in programming. It is sufficient to calculate the increments of the co-ordinates for the reference points of only one-fourth of the circle and for only a single co-ordinate axis. All the other points of the circle for both co-ordinates can be easily obtained from these computations.

The calculation of the abscissas and ordinates of the reference points of a circle (in respect to its centre), approximated by an inscribed polygon (Fig. 190), is carried out according to the formulas

$$X_n = R \cos \varphi_n$$
$$Y_n = R \sin \varphi_n$$

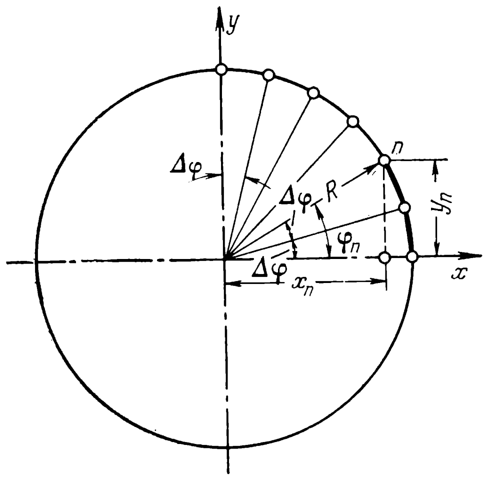


Fig. 190. Approximation of a circle:
R—radius of the circle (radius of machining); $\Delta\varphi$—angle determining the step of the approximation

where R = radius of the circle expressed in pulses

φ_n = vectorial angle of the n-th point.

The step angle $\Delta\varphi$, determining the step of the approximation, is selected on the basis of the specified accuracy and surface finish and the magnitude of the radius. In practice, $\Delta\varphi \leqslant 3°$, since with larger values of $\Delta\varphi$, clearly discernible lobed-form is observed on the machined surface.

Curves illustrated in Fig. 191 have been used for the selection of the step angle.

In programming a circle, all the blocks have identical resultant displacements along the profile (angle $\Delta\varphi$ should divide the circumference into a whole number of parts). Hence, a circle is programmed with a constant speed, i.e., with a constant time for machining each block.

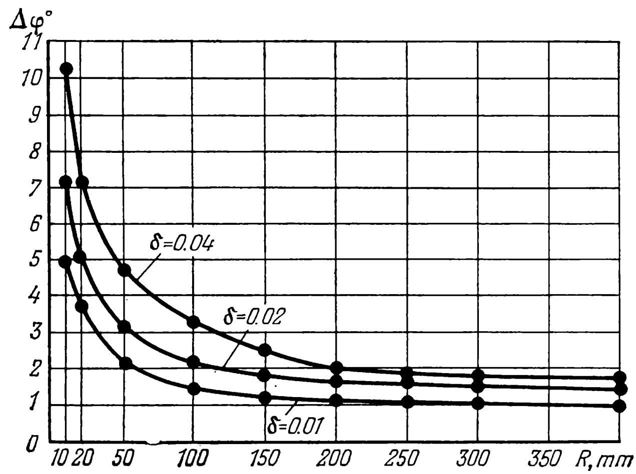


Fig. 191. Curves for selecting the step angle ($\Delta\varphi$) of approximation depending upon the radius R of machining and the permissible approximation error δ

As previously mentioned (p. 228), suitable memory units must be provided in the programme infeed unit in order to ensure uninterrupted reading of the data from the input medium, or each programme reading device should have two alternately operating tape-reading units.

14-3. Converting, Magnetic-Tape Recording and Checking the Programme

Computing devices which convert the encoded programme into a unitary code are called *interpolators*. These are decoding devices which are employed to automatically obtain the intermediate values of the co-ordinates (between the reference points) on the programmed segments to be machined. The simplest of these, so-called linear code converters, have found the widest application. In data conversion, these devices produce linearly related signals that are transmitted to control the operative units of the machine tool (along the various co-ordinates).

Some interpolators operate according to nonlinear laws, providing for displacement of the cutting tool or workpiece along circles, parabolas, ellipses, hyperbolas, etc. These interpolators, however, have a more complicated structure than linear interpolators.

The direction, path length and speed (time) of motion of the operative units of the machine tool between the reference points are specified by the programme, while the interpolator, in accordance with this programme, provides for a kinematic linkage between the operative units that is continuous within the limits of the selected discretion (pulse value). Such a linkage is very flexible; it can be easily changed by means of various numerical information fed into the interpolator. Each new block of the programme, recorded on the punched tape, may, by means of the interpolator, change the kinematic linkages between the operative units of the machine.

The principle of the decimal frequency divider of the linear code interpolator, model ЛКП-01Ф (manufactured by the Astrakhan Electronic Equipment Plant) is shown in Fig. 192. The computer proper of this converter is the setting pulse generator which continually transmits pulses of a definite frequency to a series of pulse counters, or dividers, operating on a decimally coded system (5121 code). From each divider, the pulses may be transmitted to the channels for controlling the displacements of the operative units along the co-ordinates (x and y in the diagram). The transmission of the various pulses into the circuit is determined by the contacts of the programme reading device. The closing of these contacts or any combination of them enables any amount of pulses from 1 to 9 for each decimal position (place) to be taken from the dividers (the diagram shows the dividers of only a single decimal place). Since pulses entering the control channels do not follow one another with sufficient uniformity when they are taken simul-

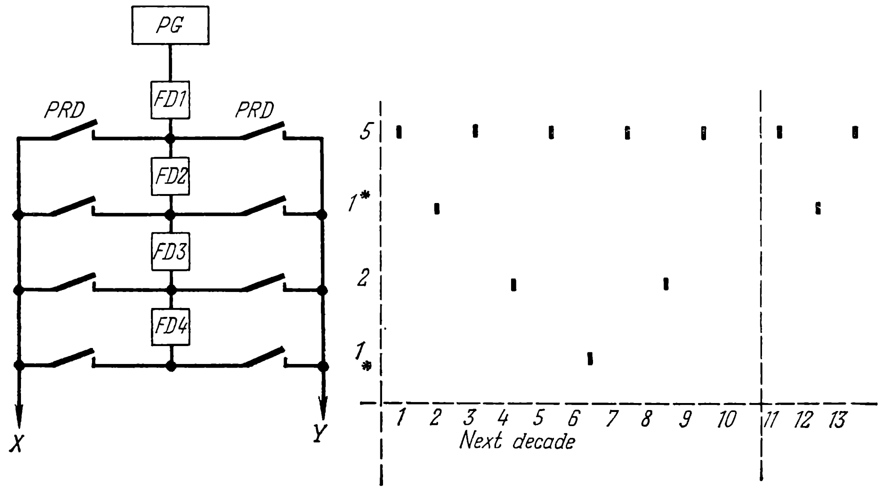


Fig. 192. Block diagram of a decimal frequency divider (the signals after each divider are shown at the right):

PG—pulse generator; *FD1*, *FD2*, *FD3* and *FD4*—frequency dividers operating on the 5121 code; *PRD*—contacts of the programme reading device

taneously from several dividers, and this may lead to errors in the motions of the machine tool units, a "hopper" is incorporated in the converter circuit. This "hopper" is a scale-of-four counter-divider. A repeat counter, providing for fourfold repetition of the sequence of all pulses taken from the dividers, prevents any changes in the number of pulses taken from the output. The application of both the "hopper" and the repeat counter ensures more uniform transmission of the pulses through the control channels without changing their total number.

For all practical purposes it may be assumed that the probable *interpolation error does not exceed the pulse value.*

The special 5121 code was selected so that the figures that are most frequently observed in machine component drawings could be more simply expressed by the code, and the succession of signals from the interpolator would be more uniform.

Decimal numbers are coded for the model ЛКП-01Ф interpolator as follows:

$0 \longrightarrow 0000$	$5 \longrightarrow 1000$
$1 \longrightarrow 0001$	$6 \longrightarrow 1001$
$2 \longrightarrow 0101$	$7 \longrightarrow 1010$
$3 \longrightarrow 0011$	$8 \longrightarrow 1011$
$4 \longrightarrow 0111$	$9 \longrightarrow 1111$

The dividers of the code converter are built up of ferro-electric transistors.

To maintain a constant rate of feed along the profile or, on the opposite, to vary it from processing considerations (reducing the rate to avoid over-travel, for instance, at sharp changes in the profile, rapid traverse in positioning, etc.), variation of the time required to transmit one block is introduced into the programme, taking into account the number of pulses. The time for transmitting the data from one block can be established by specially punched holes in the tape.

The volume of the dividers in the code converter can be reduced by 10- or 100-fold, but it should be noted that the volume of a divider cannot be reduced arbitrarily; it must be kept deliberately larger than the number of pulses taken from the divider and transmitted to the operative units for each co-ordinate.

Signals taken from the various channels of the interpolator may be fed directly into the machine tool for execution of the programme by the various operative units, or they may be recorded on magnetic tape which, in this case, is the secondary storage medium. No solution has yet been reached as to which procedure is more expedient. Both are found in various models of various manufacturers. The following arguments have been presented. An interpolator is complex, expensive, requires definite operating conditions, can process data for several machine tools and so can be more expediently installed outside of the shop. On the other hand, data recorded on magnetic tape can be easily distorted if metallic dust gets on the tape, the use of magnetic tape makes it difficult to employ cutters whose diameter deviates (for instance, after sharpening) from the design values, and intricate accurate workpieces require great volumes (lengths) of magnetic tape to record the necessary data. All of these factors dictate a different solution: to equip each machine tool with an interpolator.

Whatever the procedure resorted to, it is advisable to check the programme recorded on the punched tape. This can be done, for example, by means of a checking table (Fig. 193) which is a co-ordinatograph with slides, travelling along the co-ordinate axes and driven by *electric step motors*.

An electric step motor (Fig. 194) ensures a strictly maintained angle of rotation of the rotor when a constant voltage is applied over its windings. It is necessary to apply the voltage to the various windings (sections) of the stator in a definite sequence depending upon the required direction of rotation of the rotor. The frequency of the pulses transmitted to the step motor varies the angular velocity of rotor motion. It should be noted that step motors can be used for numerical controls in an open-loop system only within the limits of their resolving capacity.

The step motor *pickup frequency* is the instantaneous frequency differential produced by the motor without omitting even a single pulse. The resolving capacity of various Soviet step motors ranges from 80 to 1,600 cps. Upon

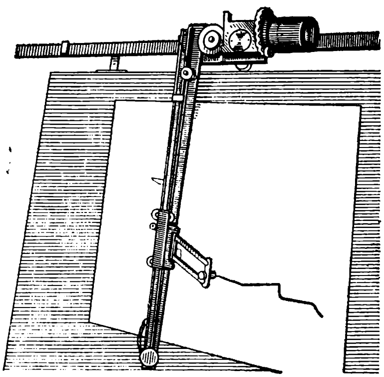


Fig. 193. Checking table (co-ordinatograph of a design based on a standard drafting machine)

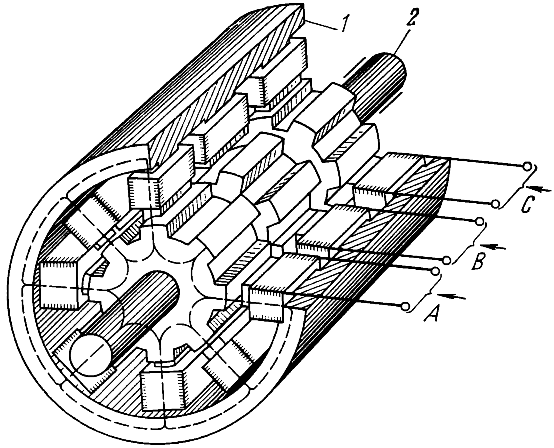


Fig. 194. Principle of an electric step motor: *1*—stator; *2*—rotor shaft; *A*, *B* and *C*—sections of the stator

smooth acceleration, these motors can operate at higher frequencies, for instance, up to 4.5 kcps. Step motors are also characterized by their dynamic torque which has a value, corresponding to the motor pickup frequency, of 2.4 to 0.5 kgf-cm (for those used as servomotors). The step at the output shaft may be as large as 6°, but 3° and 1.5° are most frequently used. Depending upon the load, the step error may reach 20 per cent of the step value, but is not accumulated in operation of the motor.

It is advisable to check the programme recorded on the punched tape or recorded on magnetic tape in the following way. Data from the punched tape are fed into the interpolator, block by block. The signals are taken off, either directly after the interpolator, or from the terminals of the magnetic head making the record on the magnetic tape, and are fed into the electric step motors used as drives actuating the slides of the co-ordinatograph.

The initial position of each step motor is noted on the corresponding scales and by indicator arrows fastened to the rotors of the step motors. In a definite scale, the recorder of the co-ordinatograph draws out the path of the programmed contour on a sheet of paper. From the reproduced drawing, it is possible to judge whether the compiled programme is correct, and the accuracy with which the step motors return to their initial position (it is advisable to compile the machining programme with a return to the initial point) is an indication of the accuracy of the recorded programme.

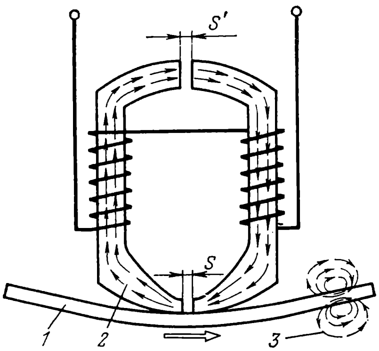


Fig. 195. Principle of a magnetic recording head:

1—magnetic tape; 2—one of the cores with windings; 3—magnetic field produced by the trace left on the tape in recording a signal; S and S'—working (front) and back gaps between the halves of the core

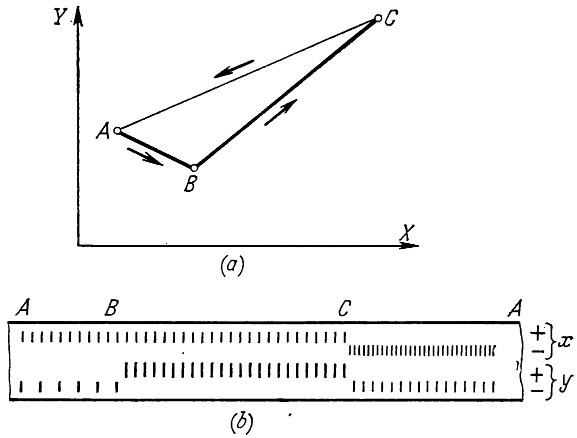


Fig. 196. Pulse recording of a programme on magnetic tape:

(a) programmed path of working motion A-B-C with rapid return to the initial point; (b) programme of motion along contour A-B-C-A recorded on magnetic tape

The magnetic tape used for recording information is a plastic base coated with a layer of ferromagnetic emulsion 20 microns thick. Use is made of tape 18.9 or 35 mm wide. The magnetic recording of the electric signals is done by means of a recording head (unit) consisting of a series of permalloy cores with windings (only one core is shown in Fig. 195, the others are behind it). The working (front) gap S between the halves 2 of the core ranges from 10 to 20 microns; the back gap S' should be as small as possible. When current is passed through the winding, a magnetic field is produced in the front gap which acts on the moving magnetic tape 1 and leaves traces in the form of magnetic lines with a magnetic field 3. The programme for the different co-ordinates is recorded simultaneously on different parallel tracks of a single tape. Reverse movements may be recorded by using a change in the polarity of the signal or on a separate track.

Figure 196 illustrates the magnetic tape recording of a programme for machining the section ABC of a profile with rapid return to point A (reverse movements are recorded on separate tracks). Actually, the magnetic lines on the tape are invisible.

In using the model ЛКП-01Ф interpolator (maximum output pulse frequency of 2.5 kcps) and a magnetic tape speed of 200 mm per sec, $\frac{2,500}{200} \simeq 13$

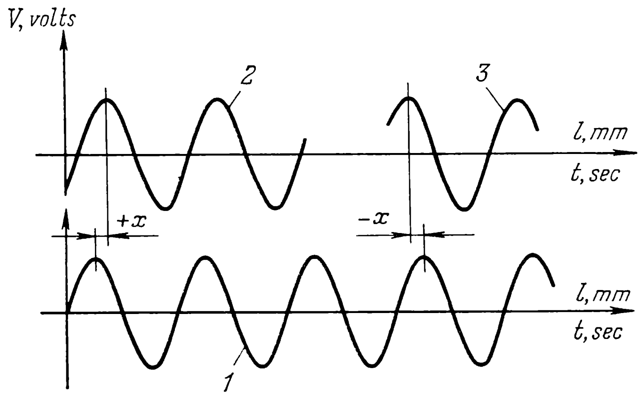


Fig. 197. Principle of the phase system of programme recording on magnetic tape: *1*—reference signal; *2* and *3*—operative signals with phase shifts of $+x$ and $-x$, respectively, in relation to the reference signal

pulses can be recorded per millimetre of tape. In practice, a lower frequency is employed, of the order of 600 cps, so that only 3 pulses per mm are recorded, or a maximum up to 6 pulses per mm. This is due to the limited capacity of the step motors for response to higher frequencies (the capacity of the magnetic tape is much higher—up to 40 pulses per mm).

The principal shortcoming of the pulse (discrete) system of executing the programme is the possibility of losing data in case command pulses are skipped (in reading the magnetic tape or the feedback signals). Another drawback is the sensitivity of such systems to pulse noises. The interference-proof quality of the system can be increased by employing a phase system of recording the programme instead of simple pulses. In this case, the so-called *reference frequency* signal is recorded on a separate track (called the reference track) while the operative frequency signals (having the same frequency as the reference signals) are recorded on the remaining tracks (as many as there are working co-ordinates). In the control system of the machine tool, the operative signal is continuously compared in phase with the reference signal, and at each moment of time the phase shift determines the displacement of the travelling unit. The phase system of data recording is illustrated in Fig. 197 (signals entering the magnetic head: reference signal *1* and operative signals *2* and *3* with phase shifts of $+x$ and $-x$, respectively).

The signals are fixed on the tracks of the magnetic tape by different degrees of magnetization of the tape (for example, with a sine-curve relationship along the tracks).

14-4. Programme Reproduction. Elements of Control Devices and Drives of the Operative Units

As previously mentioned, only the data on a comparatively small number of reference points of the contour being machined are indicated in the initial planning sheet of a contouring N/C system which contains the primary information on the machining of a workpiece. All the remaining information on the intermediate points of the contour is secondary; it is determined and fed out by the interpolator. These secondary data can be fed directly into the machine tool or recorded on magnetic tape. In the latter case, the apparatus for processing and recording the information is separated from the machine tool and installed in premises outside of the shop. In such cases, the storage medium, in the form of magnetic tape, is entered into the machine tool.

Devices resembling stationary tape-recorders are used for reading the tape. In design, the magnetic reading heads are similar to the recording heads (with several cores, depending upon the number of tracks to be read). The programme is read as the tape passes by the magnetic heads. The magnetic "lines" in pulse recording, or the nonuniform magnetization ("waves") in phase recording, produce a variable magnetic flux in the core gap which induces an e.m.f. in the windings of the magnetic reading head.

In open-loop systems of contouring controls, the programme can be reproduced by means of the electric step motors considered above or any other sources of motion providing metered displacements of constant magnitude.

The low torque developed by step motors enables them to be used in machine tools, as a rule, only as servomotors which control the power drives through suitable amplifiers. The direction of rotation of the step motor is determined either by the programme which can be recorded on the tape in the form of relatively long-time voltage pulses impressed along the tracks separately for each section of the motor with a time shift, or, in ordinary recording, by a special mark indicating the direction (by a definite polarity of the signal or by separate tracks for each direction). In the first case, the control circuit of the drive is simple but requires a large number of tracks on the magnetic tape and multiple-channel reading heads. The second form of recording is more frequently applied. It requires a special device for distributing the programme signals among the sections of the step motor in a definite order. The distribution unit is an electronic commutator assembled of transistors or (in old designs) of thyratrons, operating on the so-called ring circuit.

Closed-loop control systems must have a feedback transducer. In contouring systems, feedback transducers produce signals that correct the control process. These transducers may be of either the discrete or analog type. Their indications are read by a contactless method since they usually operate at con-

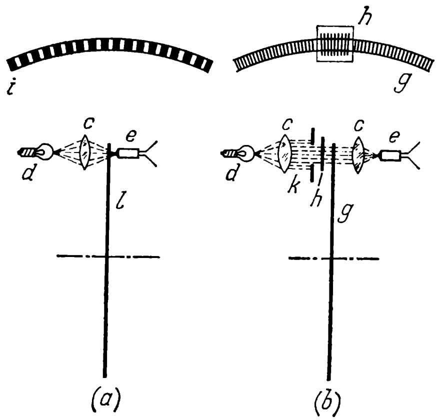


Fig. 198. Examples of circular discrete transducers:

(*a*) with radial slits; (*b*) with an additional grating *h*; *i* and *g*—disks; *e*—photoelectric cell; *c*—lens; *d*—illuminator; *k*—diaphragm

siderable speed. The control process is continuous in contouring systems. Consequently, such systems should be designed as servodriven single-counting systems. If rotary transducers are used, high requirements are made to the transmission in respect to backlash elimination. It is also important to obtain a small pulse value regardless of the type of transducer used (0.01 or 0.02 mm, and for precision machine tools, as small as 1 micron).

The simplest of the discrete transducers comprises a disk with radial slits read by photoelectric cells (Fig. 198*a*). The operative unit is usually linked to such a transducer through a rack-and-pinion drive and step-up gearing, or the transducer is mounted directly on the screw which transmits motion to the unit. In precision machine tools, the transducers are usually circular or linear scales with a small quantum value and precisely graduated divisions (the graduation accuracy may be as high as 1 or 2 microns per metre of length or several seconds of arc in 180°). The signal is initiated by the simultaneous projection of several scale graduations through an additional grating with the same graduations onto a photoelectric pickup (Fig. 198*b*). It is possible, in some cases, to reduce the quantum value of the transducer without altering the quantization of the scale itself. For this purpose, the additional grating is made with a larger number of divisions than on the main scale (but a multiple of the main scale pitch) and the signals are read each time when the next graduation of the scale coincides with one on the additional grating (vernier principle). A disadvantage of this transducer is the comparatively weak working signal since the difference between the presence and absence of a desired signal is reduced. If the scale is densely graduated, i.e., with a very small scale quantum value (of the order of thousandths of a millimetre), it will be difficult to read by the conventional method. Such being the case, the Ferranti type of transducer (Ferranti Ltd., Edinburgh) can be employed. It has transparent scales with diffraction gratings which form a Moiré fringe pattern (see Fig. 177). For this purpose, a second short grating is arranged in front of the main grating (scale) and is slightly inclined to it. Upon relative motion of these gratings, clearly visible lines are observed that move perpendicular to the scale motion. These comparatively widely spaced lines are the result of scale conversion of the scale being read. The

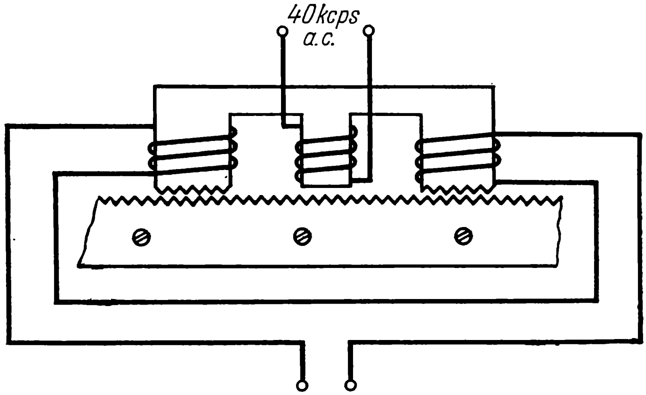


Fig. 199. Example of a linear scale read by the inductive method

distance between these lines can be readily changed by tilting the auxiliary grating.

Reading can be performed by variations in the inductivity or the phase. Here linear scales with small teeth (Fig. 199) are employed, or reading is done by special devices operating on the principle of rotary resolvers, selsyns or Inductosyns (see Fig. 178).

Information on the movement of the operative unit of the machine tool is transmitted by the feedback transducer in the form of electric signals and is fed into the comparison unit to which the input signals, determining the motion of the operative unit, are simultaneously fed from the interpolator or magnetic tape.

In case of pulse infeed of the programme, the comparison devices are usually reversible counters which sum up the pulses algebraically. A special decoder directly follows the reversible counter for converting the difference in the signals (pulses), fixed by the counter, into the proportional level of the output voltage. The counter, in comparing the programme pulses with the pulses of the feedback transducer, fixes the error in reproduction of the programme, while the decoder should have an output voltage proportional to this error. The output voltage is subsequently used for controlling the servodrives of the operative units.

Block diagrams of a *reversible counter* and a *decoder-converter* (number-to-voltage converter) are shown in Fig. 200. The counter consists of a number of cells with triggers T that actuate the gates G in their cells. The gates, in turn, prepare the cells of the higher-order places for operation. Depending upon their signs, pulses from the programme enter the addition input (input *1*) or the subtraction input (input *2*). In the zero state of the counter

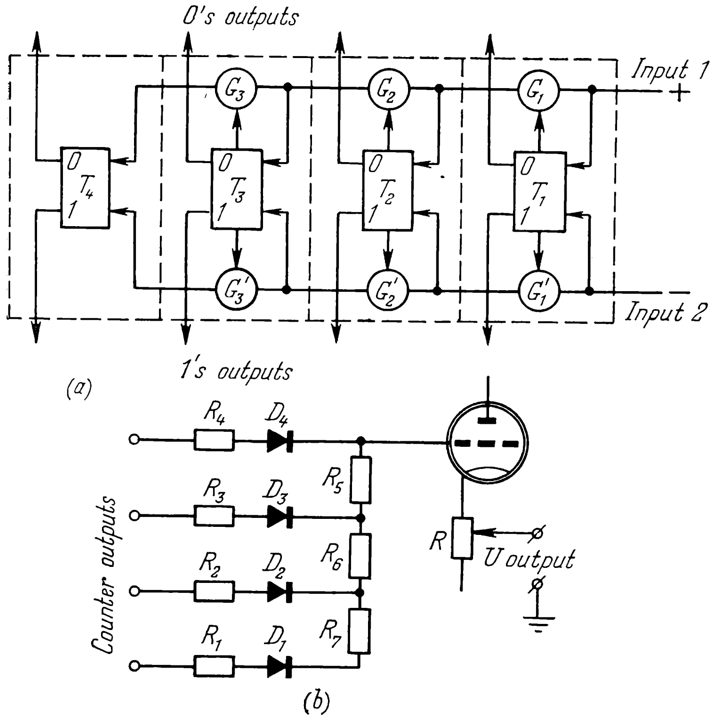


Fig. 200. Diagram of a device for pulse comparison of the programmed signals with the signals of the feedback transducer:

(a) block diagram of the reversible counter; (b) diagram of a device for converting the difference in the number of pulses into voltage; R and R_1 through R_7—resistors; D_1 through D_4—diodes

(cells in the 0000 position), gates G_1, G_2 and G_3 are closed, gates G_1', G_2' and G_3' are open and the pulse entering input *1* trips only the first trigger T_1. At this, the first cell goes over to the 1 state, gate G_1 is opened and gate G_1' is closed. The second pulse passes simultaneously through gate G_1, transferring the second cell to the 1 state and returning the first cell to the 0 state. This opens gate G_2 and closes gate G_2'. The third pulse trips trigger T_1 again and opens gate G_2'. The position of the two lower-order cells now corresponds to the 11 state. The fourth pulse changes the third cell to the 1 state and the second and first cells are changed back to the 0 state. The circuit is now in the state shown in Fig. 200*a* (corresponding to the figure 4 in the 8421 code). If in this state of the circuit, a pulse is transmitted by the feedback transducer to input *2*, triggers T_1, T_2 and T_3 are tripped and the cells will be in the

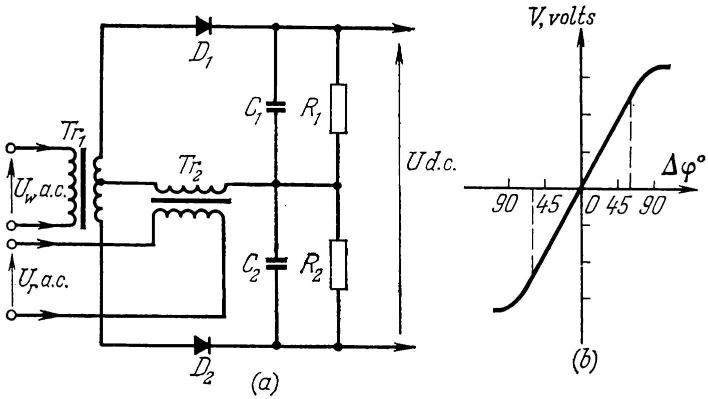


Fig. 201. Circuit diagram of a device for phase comparison of the programmed signals with the signals of the feedback transducer:

(*a*) circuit of a diode phase discriminator; (*b*) curve of output voltage variation vs phase shift angle; D_1 and D_2—diodes; C_1 and C_2—capacitors

011 position (the counter thus fixes information on the figure 3). A new negative pulse will put the counter in the 010 position, etc. It should be noted that signals from the programme and feedback transducers are always supplemented by attributes indicating the direction of motion.

To enable signals to be taken into account as they enter the addition or subtraction input in any position of the counter, the highest-order cell is put into the 1 state in the initial position of the counter. This means, for example, in a four-place counter for the binary system of notation, that the largest difference registered by the counter is 15 pulses (+7 and —8). At this, the limiting states of the counter are 1111 and 0000, since the entrance of the next signal (+8 or —9) leads to overflow of the counter.

A converter, also called a decoder, converts the information on the error in executing the programme into a signal which can actuate the drive of the operative unit. Shown in Fig. 200*b* is one version of a decoder which converts the number fixed by the counter into a voltage proportional to this number. The converter may be designed on the differential principle and consist of two chains of summating resistors, connected to the 0's and 1's outputs of the counter cells and a summing electronic amplifier. By simplifying the circuit (only one chain is shown in the figure), it can be readily conceived that various resistances will be connected to the tube grid, depending upon the state of the counter, and different voltages will be taken from the amplifier output. The circuit parameters are selected 50 that in the initial state of the counter, corresponding to the 1000 state of the cells, the output voltage of the decoder-converter is equal to zero.

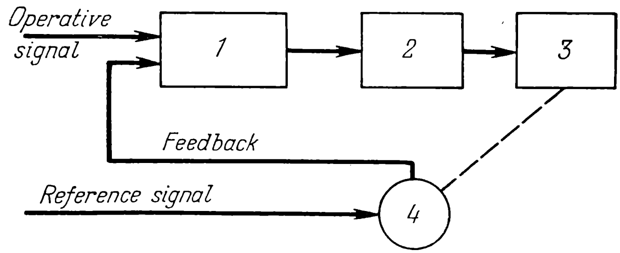


Fig. 202. Block diagram of the phase system of contouring controls:
1—phase discriminator; *2*—drive; *3*—operative unit; *4*—feedback transducer (selsyn, rotary resolver or any other transducer of the phase type)

If the phase system of executing the programme is applied, the comparison and conversion device is a unit for converting phase shift into voltage and is called a *phase discriminator*. The circuit of a diode phase discriminator is shown in Fig. 201. The sinusoidal working (primary) signal U_w from the magnetic tape, or the converted signal after the interpolator, is fed into the primary winding of a transformer while the signal U_r from the feedback transducer is fed to the primary winding of a second transformer whose secondary winding is connected to the middle point of the secondary winding of the first transformer. The output voltage U is of a value that varies with the angle of phase shift $\Delta\varphi$.

Figure 202 is a block diagram (for one co-ordinate) of the phase system of contouring controls. The phase system of executing the programme is a typical representative of analog systems. Reference sinusoidal voltage from magnetic tape or from an interpolator is fed to the stator of a rotary resolver or selsyn used as a feedback transducer. The feedback signal, whose phase depends upon the position of the transducer, is taken from the rotor of the resolver or selsyn and is transmitted to the conversion unit to which the control signal is simultaneously transmitted from the tape or interpolator. The difference in the phase of the signals is converted into voltage that controls the drive.

Selsyns can also be used in designing open-loop N/C systems. A block diagram with a selsyn used as a servodrive (the so-called synchrotie circuit) is shown in Fig. 203. In structure it resembles a circuit with an electric step motor. The magnetic tape has a track with a voltage of the carrier frequency f_0 and one with the voltage of the working frequency $f_0 \pm \Delta f$. After amplifying, the carrier frequency signal is converted into three-phase voltage and is applied to the three-phase winding of the selsyn; the working signal is applied to the single-phase winding. At this, the rotor of the selsyn runs at a speed proportional to the difference $\pm\Delta f$. Because of the low torque developed by the selsyn, it should be followed by an amplifier and a power drive.

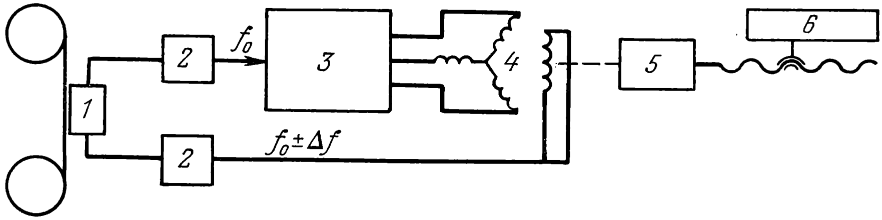


Fig. 203. Block diagram of an open-loop system of contouring controls with a phase-type servodrive (synchrotie principle):

1—magnetic head reading the programme from magnetic tape; *2*—amplifier; *3*—single-to-three-phase voltage converter; *4*—selsyn; *5*—servodrive and drive; *6*—operative unit

An important unit of devices for reproducing the programme is the drive. In contouring N/C systems, either a step drive or servodrive is used (the latter with feedback and very rarely without it, for instance, as a synchrotie). The step motor can either reproduce the movements specified in the programme or it can become a servomotor which controls a power drive. This last arrangement is more frequently applied. Step systems of numerical controls are, as a whole, open-loop control systems. However, when step motors are employed as servomotors a local closed-loop control circuit is formed between them and the power drive, operating in this case as a servosystem. Electrical or fluid-power motors are commonly used as the power drive. Since the systems of converting information in N/C machine tools have, as a rule, an electric signal at their output in the form of voltage of a definite level, it is possible, if an electrical servodrive is employed, to control it directly by means of this signal, for instance, through a rotary or other amplifier.

In cases when a hydraulic servodrive is employed, it is necessary to have a special electrohydraulic servovalve whose purpose is to convert the variable voltage, taken from the units for converting information from the programme and the feedback transducer, into control displacement of the servovalve spool. This, in turn, controls the motion of the power drive.

As an example, an electrohydraulic servovalve, type Г68-1 developed by ENIMS, is illustrated in Fig. 204. The electrical part of the servovalve consists of permanent magnet *11* with the magnetic circuit and movable control coil *8* located in the air gap of the magnetic circuit. Upon variations in the current passing through the control coil, the latter moves axially together with needle *7* secured to the coil. This occurs because in the interaction of the current in the coil winding with the permanent magnetic field a force is developed which tends to lift the coil. This force is opposed by the action of spring *9* which can be adjusted by screw *10*. In its axial movement, needle *7* can close and open the sized hole in diaphragm *6*, thereby changing the

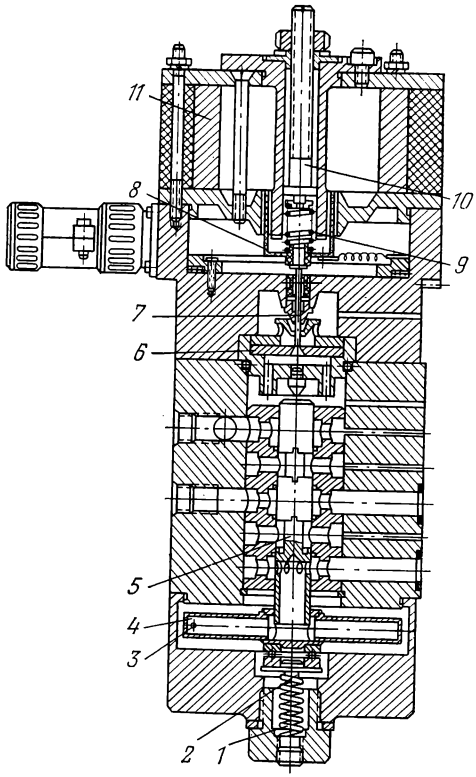


Fig. 204. Electrohydraulic servovalve, type Г68-1

pressure of liquid under the diaphragm. As the needle moves upward the oil pressure drops; as it moves downward the pressure is increased. To eliminate the influence of static friction forces, an oscillating axial motion is imparted to the needle by means of sinusoidal current at a frequency of 50 cps which comprises only a small part of the current in the control circuit. This is accomplished by the supplementary application of alternating voltage to the control coil.

The electrohydraulic servovalve has two stages or cascades. The increment of fluid pressure in the first stage (under diaphragm *6*) shifts the spool in the second stage. This changes the speed of the operative unit (spool shift is opposed by spring *1* bearing against thrust piece *2*). The spool rotates to reduce friction and the extent of the dead zone. By means of rotor *4*, the energy of the stream of liquid, admitted from the drain line through the hole in spool *5* and forced out through hole *3* in rotor *4*, rotates the spool at a speed of about 300 rpm. The spool shifts from the neutral to the extreme position in 0.06 sec (the stroke of the spool is 1.8 mm). The power of the control signal does not exceed 1 W.

Longitudinal traverse of the table and cross traverse of the saddle in the model 6M42П vertical milling machine are accomplished by means of two electrohydraulic servovalves of the type described above.

CHAPTER 15

EXAMPLES OF MACHINE TOOLS WITH CONTOURING (CONTINUOUS) N/C SYSTEMS

15-1. N/C Vertical Milling Machine, Model 6Н13ЭГ, Developed by the Gorky Milling Machine Plant

One of the N/C machine tools in Soviet industry is the three-dimensional vertical milling machine, model 6Н13ЭГ (Fig. 205), of a design based on the general-purpose milling machine, model 6Н13. The step system of numerical controls and all extra equipment for the machine were designed by ENIMS engineers. The electronic equipment, namely, the linear interpolator, model ЛКП-01Ф, programme recording console, type ПЗК, and the programme reproduction console, type ПРС, are supplied by the Astrakhan Electronic Equipment Plant.

The N/C system of the machine is of the open-loop type, without feedback transducers. The drives of the operative units (table travel in two directions, along the x and y co-ordinates, and cutter travel along the z co-ordinate) are controlled by electric step motors, type ШД-4, with a resolving capacity (motor pickup) of 800 cps at a dynamic torque of 0.02 kgf-m. These motors are linked to rotary servovalves which, in turn, control rotary hydraulic motors, type МГ-154а, that rotate screws with ball-bearing nuts for traversing the operative units. Shown in Fig. 206 is the hydraulic amplifier, type МГ18-14М, used in this milling machine. It consists of a hydraulic motor combined with a servovalve and has the following characteristics:

Working pressure, kgf per sq cm	50
Maximum torque, kgf-m	0.5
Maximum output power, W	100
Maximum speed, rpm	300

Depending upon the types of recording and reproduction consoles used, the magnetic tape is either 19 or 35 mm wide. The programme is recorded in the form of a series of consecutive pulses, i.e., in the unitary code (on six tracks) or in the form of a frequency-modulated signal. After reading the data by means of a magnetic head, the signals are transmitted to amplifiers and then to pulse shapers, which are trigger devices with one stable state. This initiates a pulse of rectangular shape of definite amplitude and length. Further, the signal enters the input of the commutator ring circuit, which distributes the signals in a definite sequence among the windings of the

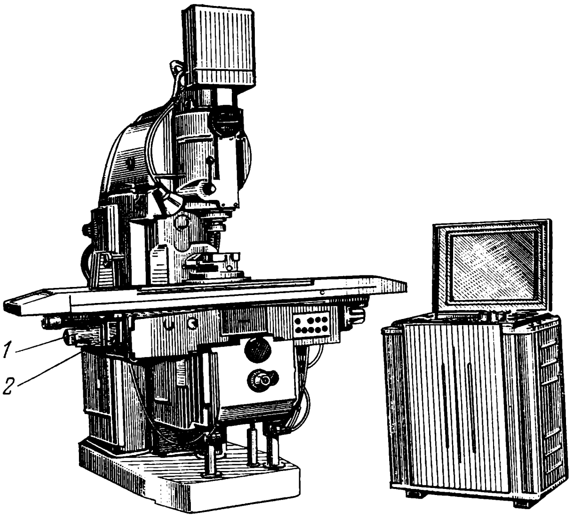


Fig. 205. Three-dimensional N/C vertical milling machine, model 6Н13ЭГ:
1—electric step motor, type ШД-4; *2*—hydraulic amplifier, type МГ18-14М

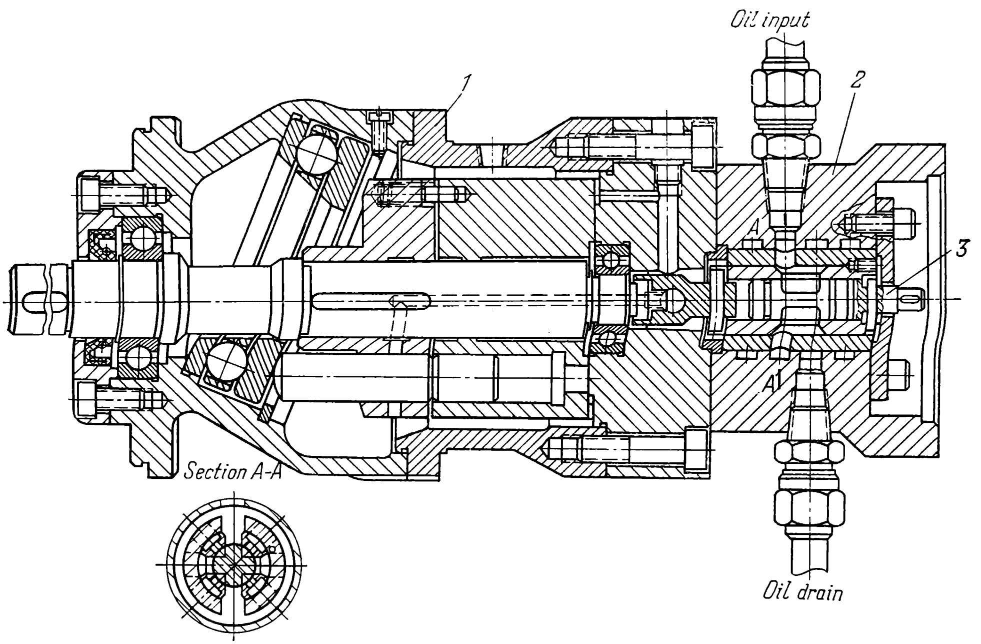


Fig. 206. Hydraulic amplifier:
1—rotary hydraulic motor; *2*—servovalve; *3*—shaft for connection to the electric step motor

electric step motor depending upon the programmed direction of travel of the operative unit.

Specifications of the Model 6Н13ЭГ Milling Machine

Table travel, mm:	
longitudinal	900
cross	285
vertical	395
Range of spindle speeds, rpm	30 to 1,500
Table feeds with numerical controls, mm per min	0 to 800
Pulse value in the control system, microns	25
Operating time according to a programme without changing the tape (tape speed 200 mm per sec), hours	1.5

15-2. Three-Dimensional N/C Milling Machine, Model 6441ПР, Manufactured by the Sverdlov Plant in Leningrad

Based in design on the tracer-controlled milling machine, model 6441Б, a three-dimensional N/C milling machine, or duplicator, was developed by one of the Moscow research institutes. Such machines are available on special order from the Sverdlov Plant of Leningrad (Fig. 207). The N/C system of this milling machine is of the closed-loop type with pulse feedback transducers in the form of disks with 524 slits read by photodiodes. A d-c electric motor, type СЛ-261, is used to eliminate backlash in the transmission from the operative unit to the feedback transducer. The motor is in the braked state and produces a constant preload in the kinematic train of the transducer drive. The accuracy of the transducer is improved by the installation of a correction bar which transmits the correction value through a lever system. The correction is made by swivelling the transducer housing.

The drives of the three operative units (for the x, y and z co-ordinates) are designed according to the servo principle using rotary amplifiers, type ЭМУ-12, and d-c motors, type МИ-32 (760 W, 2,500 rpm), for each direction of motion.

Shown in Fig. 208 is the block diagram for the numerical controls of a single co-ordinate of the machine (the operative units for the other co-ordinates are controlled by identical circuits). Input device *1* is a tape-winding mechanism with a seven-channel magnetic reading head (six tracks are used to record the programme along the three co-ordinate axes, and the seventh for checking commands). The programme signals enter unit *2* for amplification and pulse shaping. Upon movement of operative unit *7*, its motion is transmitted through a rack-and-pinion drive and reducing gear *6* to the disk of the feedback transducer *5*. Signals from two photodiodes pass

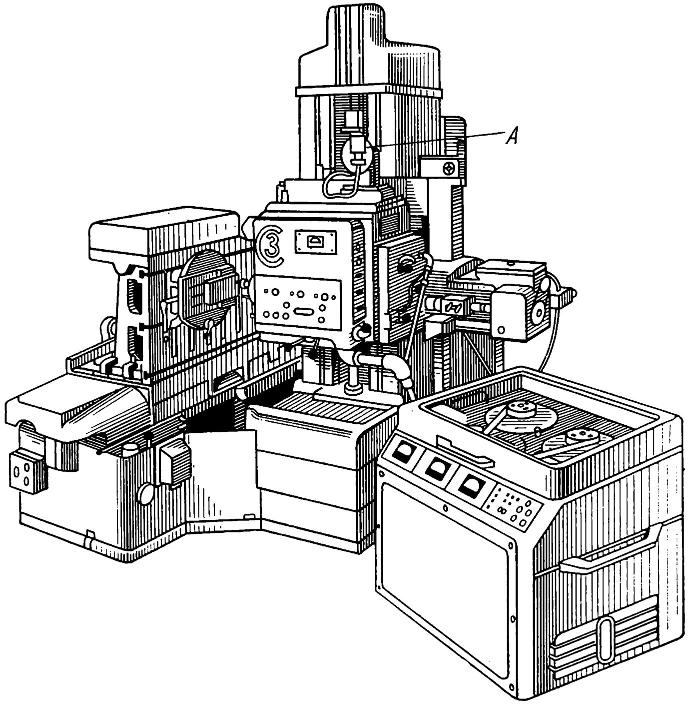


Fig. 207. Three-dimensional N/C milling machine, model 6441ПР:
A—feedback transducer

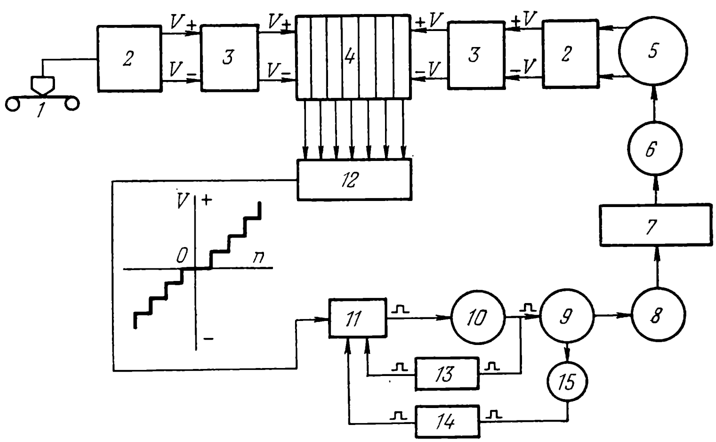


Fig. 208. Block diagram for the numerical control of a single co-ordinate in the **model** 6441ПР milling machine

through amplifiers, arranged adjacent to the feedback transducers, into the unit for shaping and distributing them into two channels depending upon the direction of movement of the operative unit.

Pulses from the infeed device and from the feedback transducer pass through synchronizing unit *3* and binary reversible counter *4*. The method of synchronization excludes simultaneous input to the counter of programme signals and feedback signals (which could lead to a loss of data).

The reversible counter, which operates as a comparison device, consists of seven positions (places), i.e., it can fix from 0 to 128 pulses. The initial ("zero") position of the counter corresponds to 64 pulses, i.e., one half of its capacity. The overflow trigger is connected to the counter output. It interlocks the servosystem if the counter overflows. The difference in the number of pulses is converted into voltage at various levels by decoder *12* (shown adjacent to the decoder is the curve for the variation of control voltage V in accordance with the number n of pulses). The obtained voltage is applied to the power amplifier *11* which controls rotary amplifier *10*. The rotary amplifier supplies d-c motor *9* which, through reducing gear *8*, powers the operative unit *7*. If the command pulse passes to the counter input through the addition channel (+), then the feedback signal passing along the subtraction channel (—) removes the command pulse. Units *13* and *14* with tachogenerator *15* are stabilizing couplings of the system.

The average machining error for workpieces of complex profile is 0.08 mm in the model 6441ПР milling machine. The labour input in machining such parts is reduced 10- to 12-fold in comparison with a tracer-controlled model, while the time required to make a part from the moment the drawings are received is reduced sevenfold, on an average.

Specifications of the Model 6441ПР Milling Machine

Size of surfaces milled, mm	900×500
Range of spindle speeds, rpm	63 to 3,150
Range of feeds of the operative units, mm per min	24 to 570
Pulse value in the control system, microns	20
Time of operation to a programme without changing the tape (tape speed 100 mm per sec), hours	2.8

TRANSFER MACHINES

CHAPTER 16

GENERAL PROBLEMS OF TRANSFER MACHINE DESIGN

16-1. Basic Conceptions

One of the chief trends of engineering progress in present-day production—integrated automation—is characterized in the engineering industries by an extensive application of transfer machines.

Transfer machines are a further development of *line production* in which the processing equipment is arranged in the order of the sequence of manufacturing operations. Production lines may be nonautomatic, semiautomatic and fully automatic.

In a semiautomatic production line, the loading of the blanks, unloading of the finished components, workpiece inspection and sometimes (in large-lot production of large-size workpieces) engaging of the handling system are not automatic. In an automatic production line, the operator loads the blanks and usually checks the workpieces, though automatic inspection is incorporated in some lines. The handling and conveying devices are automatically engaged, and the line always operates with a preset positive pace.

An *automatic transfer machine* (or simply transfer machine) is a system of metal-cutting machine tools and other processing units and handling devices which automatically carries out the processing operations in the planned sequence without the direct participation of the operator, and which requires only periodical checking, setting and maintenance from the servicing personnel.

The concept of a transfer machine intended for machining workpieces comprises the elements required for accomplishing the purpose of the machine:

(1) the workpiece in its initial state—the blank—and in the state in which it leaves the transfer machine—the finished or semifabricated part*;

(2) the sequence of operations of the manufacturing process and the corresponding tooling.

The main operative items of equipment of a transfer machine are the separate machine tools, the handling system and the control system.

* The terms "part", "component" and "workpiece" will be employed as synonyms in the following, as is general practice.

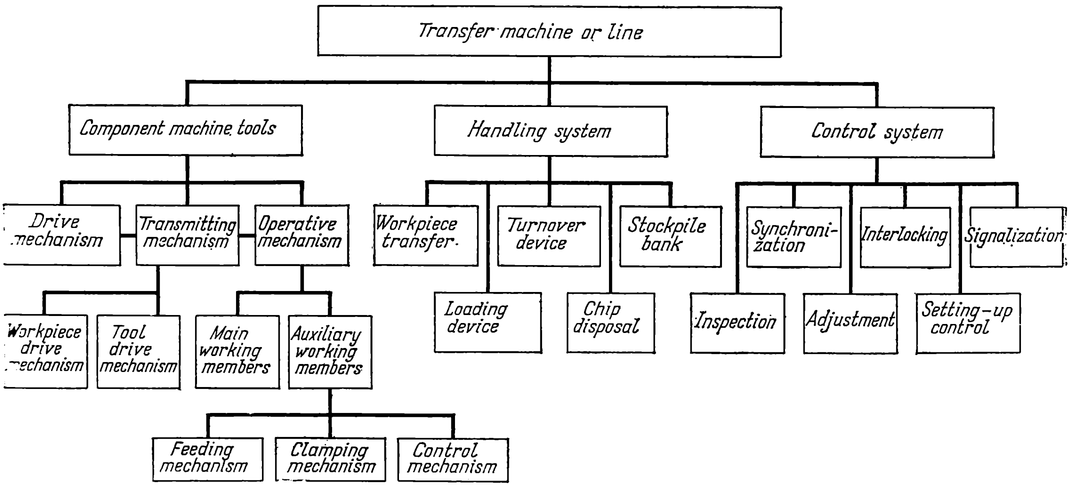


Fig. 209. General classification of component parts of a transfer machine or line

The handling system consists of the main conveying line and the following supplementary devices:

(1) banks for accumulating stockpiles of qualified parts to keep other sections of the machine in production while certain units or mechanisms are shut down for resetting, adjustments or maintenance;

(2) devices for turning the workpieces;

(3) loading devices for transferring the workpiece from the main conveying line to the clamping device of a component machine tool or to its blank feeding mechanism;

(4) automatic chip-disposal devices.

A schematic classification of the component parts of a typical transfer machine is shown in Fig. 209.

The application of transfer machines increases the productivity of the component equipment and of labour. The number of machine tools and the required floor space are reduced by 33 to 50 per cent, and the number of operators by 80 to 88 per cent. The quality of the product is considerably improved and is maintained stable. The length of the production cycle is reduced, as is the amount of in-process inventory, while the turnover of the working capital is accelerated. Machining costs are reduced by 71 to 77 per cent. At the same time, production techniques in the plant are raised to a new and more advanced level.

Drawbacks, inherent in transfer machines, include the following:

(1) Higher requirements are made to the blank so as to ensure stability of the machining process.

(2) Much time is required to change over the machine to handle a different part.

(3) The utilization factor of certain component machine tools and units may be reduced due to shutdowns of other equipment.

(4) Sometimes a component machine tool of a transfer machine has a lower production capacity than one tended individually by an operator in a production line, and therefore it is not always feasible to include all the manufacturing operations in the transfer machine (locating surfaces of housing-type parts are usually roughed beforehand in an individually handled machine tool).

(5) High-skilled servicing personnel is required.

(6) Initial expenditures are increased.

(7) Operational development and debugging may be very difficult and require a great deal of time.

Given an efficiently planned manufacturing process, a well-designed transfer machine layout, and a correct selection of the component machine tools, processing units, handling and auxiliary equipment, the advantages of a transfer machine will always far outweigh its shortcomings.

16-2. Types of Transfer Machines

Transfer machines are classified in accordance with various features. Depending upon the required production rate per hour, either *single-flow* (progressive action) transfer machines or *multiple-flow* (parallel-progressive action) transfer machines are employed. Transfer machines are divided into parallel lines of flow when the required volume of production is high and two or more machine tools are required to perform a certain processing operation.

The diversion and convergence points from single- to multiple-flow lines and back again divide transfer machines into *groups* of stations. Sometimes turnover, orienting and checking devices are introduced which also divide the transfer machine into such groups.

If stockpile banks are provided in the transfer machine, they divide it into parts called *sections*. All the machine tools and other stations within a section are interlocked, and malfunctioning of any machine tool, processing unit, element of the transfer system or control system leads to simultaneous downtime of all the machine tools and all the equipment of the section. This reduces the utilization factor of the component machine tools and therefore it is poor practice to have a large number of machine tools in each section. To raise the utilization factor it is necessary to reduce the number of machine tools operating in sequence to one in each section, i.e., to provide an individual stock bank before each machine tool. In this connection, transfer machines may have rigid interlinkage (interlocked transfer machines

for housing-type workpieces) or flexible interlinkage between the component stations (transfer lines for small parts, such as bushes, rollers, pins, collars, dowels, etc.).

In construction, rigid, or positive, interlinkage between the component stations is accomplished by a common transfer-bar conveying unit which establishes a general output pace for the whole transfer machine. Flexible interlinkage between the component stations can be illustrated by transfer lines in which the workpieces leaving one machine tool are delivered to a hopper or magazine of the next one. Thus, malfunctioning of one machine tool does not necessitate a shutdown in any of the other machines.

As to the types of their component machine tools, distinction is made between transfer machines made up of machine tools especially designed for the given case; of general-purpose automatic and semiautomatic machine tools; of unit-built machine tools or of automated universal machine tools (see Sec. 16-6).

In respect to the intermachine transfer facilities, transfer machines may be classified as:

(1) the pass-through type in which the workpiece passes through the clamping zone (used for machining housing-type parts in unit-built machine tools);

(2) the overhead type with the workpieces conveyed horizontally in the longitudinal direction and vertically in the transverse direction, thereby offering easier access to the component machine tools for servicing, but requiring a more complex handling system;

(3) the side-loading (frontal) type with longitudinal and transverse conveying motions;

(4) the combined-transfer type;

(5) the rotary-index or centre-column type with rotary conveying means.

The choice of a conveying method depends upon the type of workpiece to be handled, its shape, weight and other features; the nature of the manufacturing process, and the machine tool layout.

As to the arrangement of the equipment, closed-loop and open-loop transfer lines are distinguished.

Closed-loop transfer machines may be circular (with a small number of stations, using a rotary indexing table) or they may be rectangular.

Blanks are loaded into such a closed-loop transfer machine and the finished parts are removed at the same position by a single operator, this being an advantage of this arrangement. On the other hand, access to the equipment for servicing may be more difficult in a closed-loop transfer machine.

Most transfer machines and lines have an open-loop arrangement which may be straight-line (in-line), L-shaped, U-shaped, W-shaped or zigzag.

The choice of an arrangement depends upon the conditions on which the transfer machine layout is to be based.

An in-line arrangement ensures convenient access to the processing equipment. It is the first choice for a single-flow transfer machine, permits housing-type parts to be machined from two sides, and requires part turnover devices if machining is to be performed from the other sides as well. An L-shaped arrangement has the same properties but also permits a housing-type part to be machined from all sides without introducing a special turnover device.

If the available length of the shop is less than that required for an in-line arrangement, the line is folded back on itself, constituting a U-shaped arrangement. This arrangement is also used for a double-flow transfer machine or line. What we have called a W-shaped arrangement, which actually is more like a three-prong fork, is used for a triple-flow line. It extends over a group or section of the transfer machine or line between the stockpile banks.

Complex zigzag arrangements are obtained in combinations with stockpile banks for housing-type parts, located square to the sections of the transfer machine.

The most efficient transfer machine layout depends primarily upon the type of part to be machined and the manufacturing process, since these two factors determine the choice of machine tools and, together with the accepted types of machine tools, are of decisive importance in selecting the handling system. This system depends, not only on the workpiece, but also on the layout of the machine tool (see Chap. 6).

The above-mentioned elements of a transfer machine or line, including the handling system, govern not only the choice of the type of transfer machine but also its layout.

At the present-day state of transfer machine theory, any consideration of the effect of the various factors enumerated above on the choice of the elements making up a transfer machine is subjective to a great extent. This may lead to different possible solutions whose actual efficiency can be properly evaluated only as a result of a sufficiently long period of operation in regular production.

As more and more transfer machines are designed and manufactured, there is a greater diversity of workpieces that they handle. Hence, the primary classification of transfer machines (according to the type of workpiece they machine), though it is the most well-founded one, cannot be exhaustive, because it cannot embrace all kinds of workpieces and all types of transfer machines.

As to the kind of workpiece they handle, transfer machines and lines are classified as those: (1) for housing-type parts; (2) for shafts; (3) for disk-type parts (gears, etc.); (4) for ball-bearing rings; and (5) for small parts (screws, pins, rollers, etc.).

16-3. Product Design for Transfer Machine Automation

The following requirements are made to workpieces that are to be machined in transfer machines or lines:

1. It is of great importance that the construction of the part remains stable (unchanged) for a period of time sufficient to justify the cost of the transfer machine since the expenditures for the building and development of a transfer machine are usually very large.

2. The volume of production should be large enough for the costs of automation to be justified by the reduction in manufacturing costs when the parts are made in a transfer machine.

3. The producibility of the part should be such that a simple standard manufacturing process, which readily lends itself to automation, can be employed.

4. The highest economic effect can be gained with parts having a high relative share of production expenditures (wages and overhead) in the total cost of the part.

5. Housing-type parts are co-ordinated in their location for machining in respect to the datum surfaces by means of three points on these surfaces. Locating datum surfaces should be protected against chips and dirt. To enable the part to be automatically fixed in the working position by two taper pins, it should reliably retain its position (orientation) in being transferred from one station to the next.

In co-ordinating a part in space and locating it from two datum points, its accurate orientation (for shafts, disks, rings and collars) in respect to the clamping device (jaws, collet, centres, etc.) is of importance at the end of the transfer to the machine tool. In the transfer process, it is necessary to retain orientation in respect to the surfaces of the handling device.

6. The field of scatter of material hardness values should be narrower for transfer machine workpieces than ordinarily. This is required to obtain a stable tool life so that tool resetting and change schedules can be established.

7. The construction of the part should be developed to comply with processing requirements as to location and machining. Rough blanks should have the minimum feasible machining allowances. The size and machinability of the blanks should be as stable as possible.

If deviations in size of the blanks may lead to loss of orientation, jamming or incomplete clamping in fixtures, or to other malfunctioning, 100-per-cent inspection of the blanks must be planned for at the input to the transfer machine, in respect to errors in dimensions leading to trouble and breakdowns, in order to ensure trouble-free operation of the transfer machine or line.

16-4. The Manufacturing Process

One of the main criteria of the production capacity of an automatic machine tool, the output factor or, as it is called by Academician V. Dikushin, the degree of continuity of the manufacturing process (see Sec. 7-1), is also valid for transfer machines.

One of the main guiding principles in planning the manufacturing process and in designing the transfer machine or line is a maximum possible reduction of cyclic and noncyclic time losses which are not overlapped by the main processing time. From this follows the requirement that there be the minimum possible number of relocations and reclampings of the workpiece, since the time for these operations cannot in most cases be overlapped by processing time. A reduction of such operations also reduces the required amount of auxiliary handling and loading facilities. Most promising in this respect are continuous-flow transfer lines, for example a transfer line for grinding small cylindrical parts and consisting of centreless grinders in which the conveying and processing motions are combined, and the workpiece axis is fixed by the action of the cutting forces (Fig. 210).

Minimum possibilities for combining the time for transfer movements and the noncyclic losses with the main processing time and with each other are found in transfer machines for housing-type parts with positive interlinkage between the machine tools. These are interlocked transfer machines in which the transfer movements, cutting motions, idle travel and handling movements take place consecutively, being combined only in respect to the various component machine tools of a section. In these interlocked transfer machines, the design pace τ of the machine, i.e., the interval of time elapsing between the output of successive finished workpieces, is

$$\tau = t_m + t_i + t_t + t_c + t_r \tag{75}$$

where t_m = machining time, equal to the cutting time plus the dwell time of the power unit against a positive stop

t_i = time required for idle travel movements of the power unit

t_t = time required to transfer the workpiece from station to station

t_c = time for clamping and locking (if provided for)

t_r = time for releasing (unclamping) the workpiece.

In transfer machining, in addition to the usual requirements made to the choice of the locating datum in planning the manufacturing process, it is necessary to make provision for convenient handling, the possibility of automatically fixing the workpiece and the minimum number of relocating operations.

An increase in the number of cutting tools at each station enables the number of component machine tools to be reduced in the transfer machine. However, the concentration of tools at the various stations should not be so

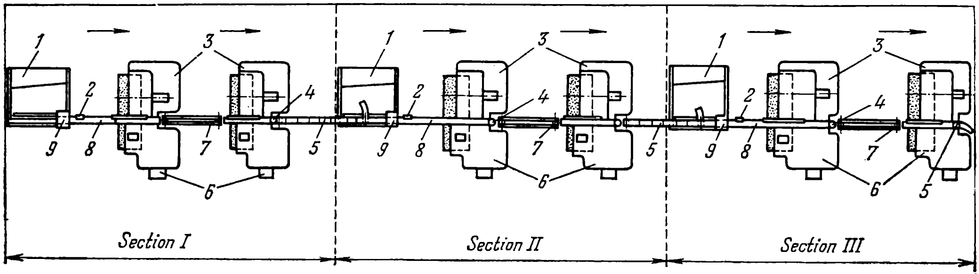


Fig. 210. Transfer line for grinding wristpins:
1—hoppers; 2—guide disks for orienting the wristpins; 3—centreless grinders; 4—automatic inspection units; 5—elevators; 6—automatic feedback size adjustment; 7—conveyers; 8—chutes; 9—distributors

great as to complicate setting up and adjustment of the tools, chip disposal and application of cutting fluid.

One essential problem in process planning is to achieve, if possible, equal times of operation at each station, or at least a time for certain operations which is a multiple of the time for the other operations. This equalizing of the times, called balancing out the transfer line, is needed to properly synchronize the operation of the various machine tools and stations.

Various methods are used to synchronize workpiece machining in the different operations.

The limiting (longest) operation is spread over several stations by dividing up the whole working travel into several parts. This is feasible for roughing operations and for drilling low-accuracy holes which are made stepped with diameter intervals of about 0.2 mm.

Certain short operations (as in machining small holes) can sometimes be combined by the use of multiple-diameter twist or core drills or other combination tools. In some cases synchrony in machining is achieved by distributing the processing operations into sections (milling, boring, drilling, etc.).

Illustrated in Fig. 211 is the sequence of operations for machining the holes on the end faces of cylinder blocks for automobile engines. The block has an oil hole which is so long that, in addition to drilling from both ends, the length of working travel is divided into six parts on one end and into seven on the other, and drilling is performed consecutively in seven working stations.

Such a division of the length of travel for limiting operations into several parts is especially expedient in drilling and boring in multiple-spindle machines, since the additional spindles can be arranged partly or wholly in the spindle heads together with the other spindles.

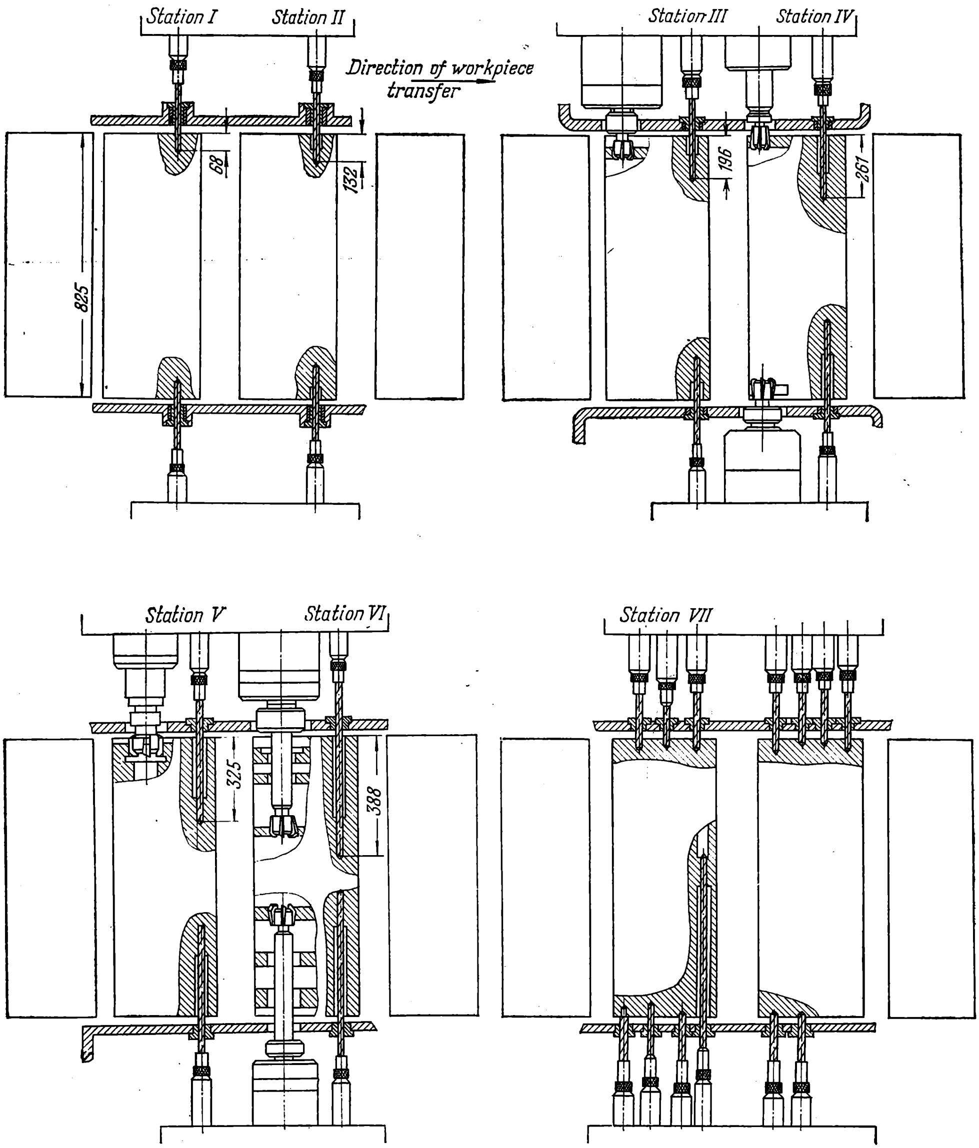


Fig. 211. Sequence of operations for machining holes in the end faces of automobile engine cylinder blocks

If it is impossible to achieve synchronized machining at a certain station, the number of workpieces simultaneously machined at the given station is increased to maintain the pace of the line, or several identical machine tools are installed at a station for an operation requiring a time much longer than the accepted pace.

16-5. Cutting Speeds and Feeds, and the Cutting Tool

Any increase in the number of simultaneously operating tools reduces the machining time and the wages per workpiece, and increases the relative share of tool expenditures. Therefore, when the number of spindles and working stations tended by a single operator is increased, optimum cutting speeds and feeds are decreased. An unwarranted increase in cutting speed and feed reduces the dependability of transfer machine operation and may lead to prolonged downtime for tool changing. Hence, in the limiting operations, feeds and speeds are assigned on the basis of a tool life of at least 7 hours between grinds, so that the tools can be changed between shifts. The time between grinds is reduced in some cases to 3.5 hours for limiting operations. Then tool changing is to be performed during the lunch interval.

If there is a large number of tools in a transfer machine, reaching 1,000 or more in a single automatic system, tool changing based only on the judgement of the setter-up frequently results in excessive dulling of some tools, breakage and an increase in the downtime of the machine.

Most of the operations performed in transfer machines do not require very high accuracy, and tool changing usually arises from a loss in cutting capacity. Since under conditions of transfer machining the cutting life of a tool does not vary greatly, unlike the size life (because of the higher requirements as to the stability of the quality of the workpiece material and its machinability), it is possible to change tools according to an established schedule, after definite intervals. This enables the time required for tool resetting and adjustment to be reduced, and tool breakage due to excessive dulling because of an oversight of the setter-up to be avoided.

To draw up a tool-change schedule, the tools are divided into groups with approximately equal life (see p. 337). In accordance with the cutting speed and feed, the tool life T in min is determined as well as the scheduled time T_s of transfer machine operation between resharpenings:

$$T_s = \frac{T}{\lambda t_m} t_c \tag{76}$$

or the number z of workpieces machined by the given tool between resharpenings is determined:

$$z = \frac{T_s}{t_c} = \frac{T}{\lambda t_m} \tag{77}$$

where λ = ratio of the actual cutting time to the machining time t_m, i.e., to the time of tool travel at the rate of working feed plus the dwell time of the power unit

t_c = time for one working cycle, min.

Cutting tools are divided into groups as follows.

With a 7-hour working shift and tool changing planned for nonoperational time (between shifts and at lunch time), tools with a scheduled time $T_s < 210$ min $\left(T_s < 3\frac{1}{2}\text{ hours}\right)$ are assigned to a special group and are changed during operational time.

Tools with a scheduled time $T_s > 210$ min are divided into groups so that the first group includes tools with $210 < T_s \leqslant 210 \times 2$ min; the second group contains tools with $210 \times 2 < T_s \leqslant 210 \times 3$ min, and the k group contains tools with $210k < T_s \leqslant 210\,(k+1)$ min.

It is considered permissible to change tools of group k beginning with the moment they have operated $0.9 \times 210k$ min, i.e., after a minimum number of cycles R_{min} of operation equal to

$$R_{min} = \frac{0.9 \times 210k}{t_c} \tag{78}$$

The maximum permissible number of cycles of operation for tools of group k is taken equal to

$$R_{max} = \frac{T_{s\,min}}{t_c} \tag{79}$$

where $T_{s\,min}$ is the minimum scheduled time between tool changes for tools of group k.

A cycle counter is installed in the tool cabinet at the place where tools of the given group are stored. After counting off R_{min} cycles, a pilot lamp lights up on the control desk; after counting off R_{max} cycles, the corresponding section of the transfer machine or line is switched off automatically.

In the accurate machining of workpieces, when the time for tool changing is determined by the size life (the time a tool will continue to cut to size), which varies in a very wide range in practice, scheduled tool changing is unsuitable. In such cases, transfer machines should incorporate devices for checking the accurate surfaces whose size depends upon the condition of the cutting edges on the tools.

Scheduled tool changing does not exclude the necessity for reducing time lost in tool adjustments by the application of quick-change tool clamping and preset tools.

The quick-change sleeve (Fig. 212), designed in Special Design Office for Transfer Machines and Unit-Built Machine Tools for shank-type tools, is adjusted axially in the spindle by means of nut *5* so as to properly align

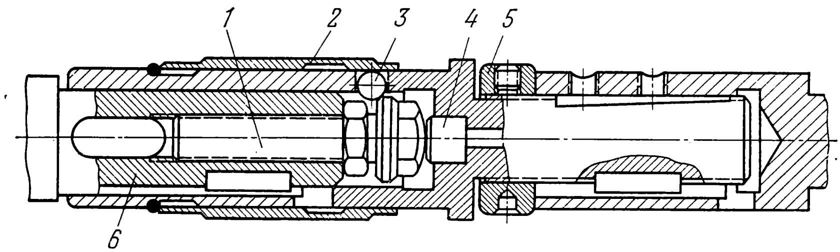


Fig. 212. Quick-change sleeve

the thrust members *4* for all the spindles of the gearbox. Adapter *6*, carrying the tool, is inserted into the body of the sleeve. Adjusting screw *1* bears against thrust member *4*. The adapter is locked in the sleeve body by means of ball *3* when bushing *2* is shifted axially by hand. The axial position of the cutting tool is set up in a setup gauge outside of the transfer machine by adjusting screw *1*.

The cutting tools are stored inserted in their adapters *6*. After resharpening, the adapter with the tool is inserted into bushing *2* of the setup gauge (Fig. 213) for which purpose lug *1* is turned to the side. Then, by means of the adjusting screw of the adapter, the tool is adjusted in length until the drill point contacts lug *1* and the head of the adapter adjusting screw contacts the tip of screw *3*.

Setup gauges of this type are employed for settings of an accuracy within ± 0.25 mm. Limit gauges are used for more accurate settings, while for an accuracy within 0.1 mm, dial indicator gauges may be used.

Quick-change preset tools are extensively used in transfer machines (see Fig. 131).

Single-point tools with mechanically clamped button or polyhedral throw-away inserts of cemented carbide (Fig. 214) are being efficiently used in transfer machines. Button insert *4* (Fig. 214*a*) is clamped by rod *3* and is held tightly against shank *1* of the tool by spring *2*. A button insert can be turned by hand, without removing the tool from the machine, to present a new sharp section of the cutting edge to the work. The adjusting screw in the shank is intended for setting up the length of tool overhang in a special fixture outside of the machine.

A polyhedral cemented-carbide insert *2* (Fig. 214*b*) fits on pin *3* with a clearance of 0.1 to 0.15 mm. Pin *3* is press-fitted in tool shank *1*. The insert is held firmly against the seat on shank *1* by means of wedge *4* and additionally by the cutting force. Wedge *4* must be released to turn the insert for changing the cutting edge.

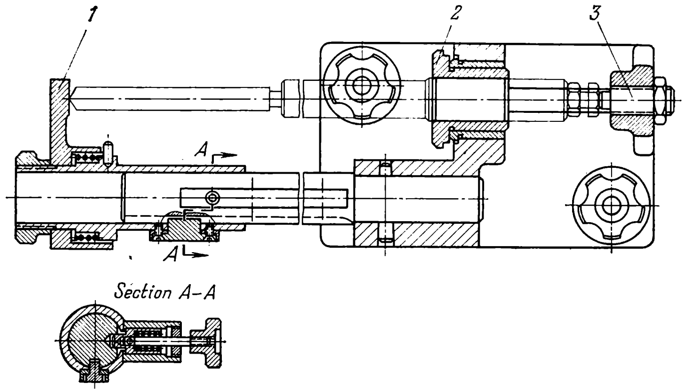
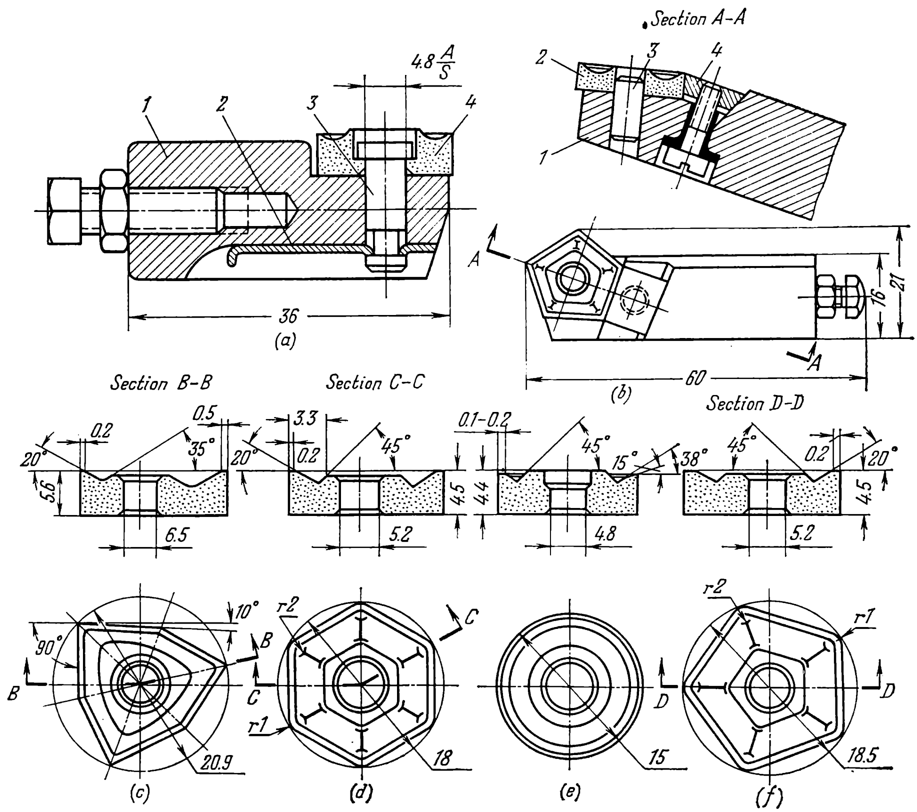


Fig. 213. Gauge for presetting the length of shank-type tools



Other types of cemented-carbide throw-away inserts (Fig. 214*c* through *f*) for tools used in transfer machines have a land 0.2 to 0.5 mm wide on the face and a chip-breaker groove which are formed in the process of manufacture of the insert.

16-6. Equipment of Transfer Machines

1. The layout of the component machine tools, especially their working zones, should provide for the application of all kinds of loading devices and transfer systems that are normally used in transfer machines for parts which are machined in machine tools of the given type.

The possibility of using a loading and transfer system of only one kind may be satisfactory if a machine tool of an existing model is built into the transfer machine or line. Comprehensive correlation of the layout of a machine tool being designed with all standard loading and transfer systems enables the machine tool to be built into a section with machine tools of other types and for other purposes. It increases its degree of versatility and extends its field of application.

This, in turn, leads to more efficient operational development of the design of the machine tool and increases its dependability. An example of such correlation is evident from a comparison of the working zones of the single-spindle semiautomatic lathes, models 1730 (Fig. 108, Vol. 1) and 1712 (Fig. 129).

The correlation of the machine tool layout with standard types of loading and transfer systems is manifest not only in the convenience of using these systems, but in the accessibility of the cutting zone (when the loading and transfer system is installed) for visual observation of chip disposal (especially in the case of continuous chips) to the chip conveyer and for eliminating factors impeding chip disposal, accessibility of the chip conveyer to remedy its jamming and other troubles, accessibility of the hydraulic apparatus and coolant system for changing packing seals, etc.

2. The layout of the machine tool, the construction of its slides, housings and bed should be such as to ensure reliable chip disposal from the cutting zone, while the chip conveyer should reliably carry the chips out of the machine tool.

Especial difficulty is encountered in disposing of long continuous chips formed in turning steel blanks in horizontal single-spindle and vertical multiple-spindle semiautomatics whose overall size conditions provide enough space for designing highly rigid spindle units capable of roughing steel blanks with carbide tools.

Best conditions are offered for continuous chip disposal from the cutting zone if, in a horizontal single-spindle semiautomatic, the longitudinal carriage with the tool is turned 180° about the line of centres. This leaves

a free space, allowing the chip to drop down without entangling the workpiece or the tooling on the carriage (see Figs. 127 and 128).

In turning stepped shafts in a tracer-controlled semiautomatic lathe, the operation of the chip breakers is impeded by the variations in the cutting conditions (rate of cutting speed and depth of cut).

Chip disposal troubles increase with the chip cross section. For this reason, it is easier to dispose of chips in transfer machines comprising grinders and small-size automatic bar and chucking lathes than in multiple-tool and tracer-controlled semiautomatics.

In transfer machines consisting of unit-built machine tools and designed for machining housing-type parts, chip disposal is not directly dependent upon the layout and construction of the component machine tools; it does, however, affect the construction of the fixture and handling devices, requiring the provision of openings for the chips to fall through.

3. The most important requirement made to the component machine tools and other equipment of transfer machines or lines is high dependability in operation. High dependability of a transfer machine is achieved by raising the inherent dependability of each machine tool or other unit, and by reducing its loss of working time due to downtime of other machine tools or other equipment of the transfer machine linked with the operation of the given machine tool.

In respect to the interlinkage between the component machine tools, distinction is made between: (1) transfer machines whose components are rigidly interlinked by a common transfer bar system which simultaneously transfers all the workpieces from station to station, the transfer pitch being equal to the distance between adjacent machine tools (interlocked transfer machines or lines); and (2) transfer machines whose components are flexibly interlinked by intermachine stockpile banks (hoppers) which allow each component machine tool to operate regardless of stoppages in the preceding or subsequent machine tool of the line.

In an interlocked transfer machine, a shutdown of any machine tool leads to downtime of the whole section having a common transfer bar system. Therefore, increased dependability of the component machine tools and other equipment of a transfer machine is especially important when they are rigidly interlinked.

One means of increasing the dependability of a transfer machine is to raise the production capacity of the component machine tools (without impairing their own dependability). This enables the number of machine tools to be reduced, together with other equipment, and thereby reduces the number of elements that may be the cause of downtime.

Of essential importance in raising the inherent dependability of component machine tools and their economic features is the application in transfer machines of lot-produced machine tools whose construction has been tested

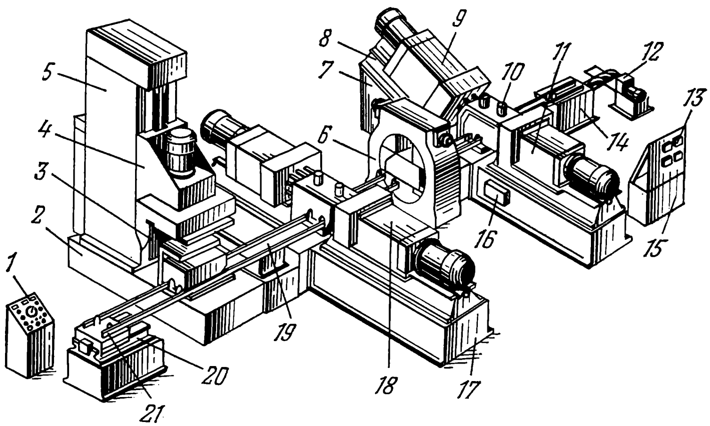


Fig. 215. Schematic diagram of a section of an interlocked transfer machine designed for machining housing-type parts.

Standard units of the transfer machine: *1*—control desk; *6*—workpiece turnover drum; *12*—chip conveyer drive; *13*—flush-mounted hydraulic apparatus; *15*—hydraulic power unit; *16*—automatic-lubrication pump; *19*—indexing table; *20*—workpiece transfer unit. Standard units of the machine tools: *4*—separate-feed power unit; *5*—column; *8*—slideway; *9*, *11* and *18*—self-contained power units; *10*—workpiece clamping cylinder; *17*—bed. Nonstandard units of the transfer machine: *2*, *14* and *21*—beds and bases; *3*—fixture; *7*—inclined bed

and developed in the course of regular operation. A characteristic feature of such machine tools is their versatility, their adaptability for being conveniently built into a transfer machine having any of the main types of loading and transfer systems normally used in machining workpieces of the given kind. Other advantageous features of these machine tools are their convenience in manual or crane loading, accessibility of the main members for setting up and adjustments and their convenient manipulation from the workplace, and convenient control of the manual and semiautomatic cycles in operation outside of a transfer machine (for example, the tracer-controlled semiautomatic lathes, models 1722 and 1712).

Milling, boring, drilling and thread-cutting operations are performed in interlocked transfer machines that handle housing-type parts. These operations are done by special high-production multiple-spindle component machine tools.

The dependability of special machine tools is raised and their cost is reduced by designing them on the unit-built principle of standard units whose construction has been tested in operation.

For the same purpose, the units of the transfer devices and control systems for interlocked transfer machines (Fig. 215) are being standardized to a greater and greater extent.

CHAPTER 17
TRANSFER MACHINES FOR HOUSING-TYPE PARTS

17-1. Features of the Manufacturing Process

Operations in which foundry defects, such as blowholes and others, are revealed should be performed as near as possible to the beginning of the transfer machine. All the large surfaces should be roughed in the initial operations to avoid distortion of the workpiece following the finishing operations which it is desirable to arrange as near as possible to the end of the transfer machine.

The number of turnover devices can be kept at a minimum if all operations for machining in a given position are completed before turning the workpiece over for machining in another position. This may be opposed, however, by certain processing considerations, such as the possibility of warping of the workpiece if it is roughed in the new position.

It proves expedient in some cases to incorporate additional turnover devices. Such devices (resembling the car dumpers used in steel plants) turn the workpiece completely over allowing the chips to drop out of drilled holes. This is necessary to prevent tap breakage in a subsequent tapping operation. Holes with interrelated locations should be drilled at the same station. Through holes are more accurately located on the side the drill enters the workpiece. Therefore, holes for fasteners should be drilled from the jointing surface of the part.

Combination assembled (built-up) tools may be employed to reduce the number of working stations and to improve machining accuracy. These include gangs of side milling cutters on arbors, boring bars with sets of cutters, boring heads or reamers. Because of their high cost and the difficulties encountered in resharpening, solid combination tools are used only in exceptional cases, such as in machining parts of aluminium alloys when tool life is very long, and in drilling and reaming locating holes when the blank is located on a rough surface thereby excluding the possibility of accurately locating it a second time for reaming the hole after drilling.

It is desirable to arrange for all tapping operations in a separate section of the transfer machine.

Rotary-table and drum-type milling machines for continuous milling are not employed in transfer machines because of the unwieldy and expensive automatic loading devices they require. They can be efficiently used,

when tended by a separate operator, for milling the locating datum surfaces of the blank prior to its being loaded into the transfer machine.

Machines with two-spindle milling heads on each side are used in transfer machines instead of rotary-table milling machines. These heads travel along ways arranged between two transfer conveyers for housing-type parts. In the working travel at the rate of milling feed, to a length equal to the length of the machined surface plus the cutter diameter, four workpieces are milled simultaneously, two are roughed and two are finish milled (Fig. 216). During the rapid return stroke, the milling head, by means of two bars with ratchet (disappearing) pawls attached to the head, transfers the workpieces in both lines one pitch (equal to the length of working travel) from the roughing station into the clamping device for finish milling. At the same time, the workpieces in all the working and idle stations of section *I* of transfer machine, model Л-52 (Figs. 217 and 218), are transferred to the next stations. On the turntable the workpiece is turned and its two other sides are milled in another milling machine of the same type. From the milling machine the workpieces are transferred to idle stations and then to a transverse conveyer.

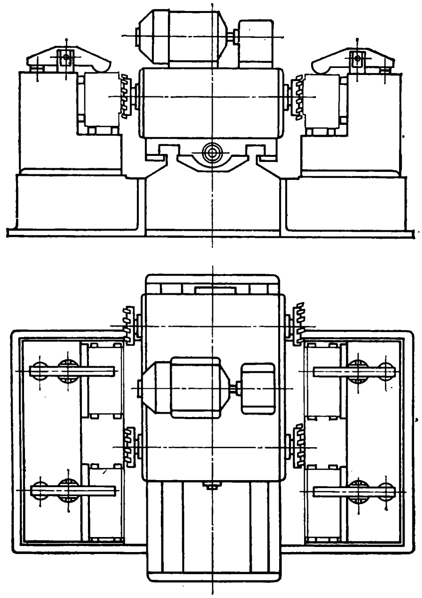


Fig. 216. Machine for milling four blanks simultaneously in two lines

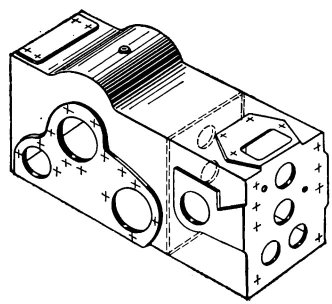


Fig. 217. Tractor gearbox housing

If several surfaces requiring a long working travel are to be milled on a stationarily clamped workpiece (Fig. 219*a*), two feed mechanisms will be needed for the two milling heads instead of one for the table if a planer-type milling machine were used (Fig. 219*b*). This makes milling machines of the first type less reliable than planer-type millers. If the workpiece slides along bars and can be clamped from the sides, planer-type milling machines permit in-line transfer of the workpieces through the clamping position,

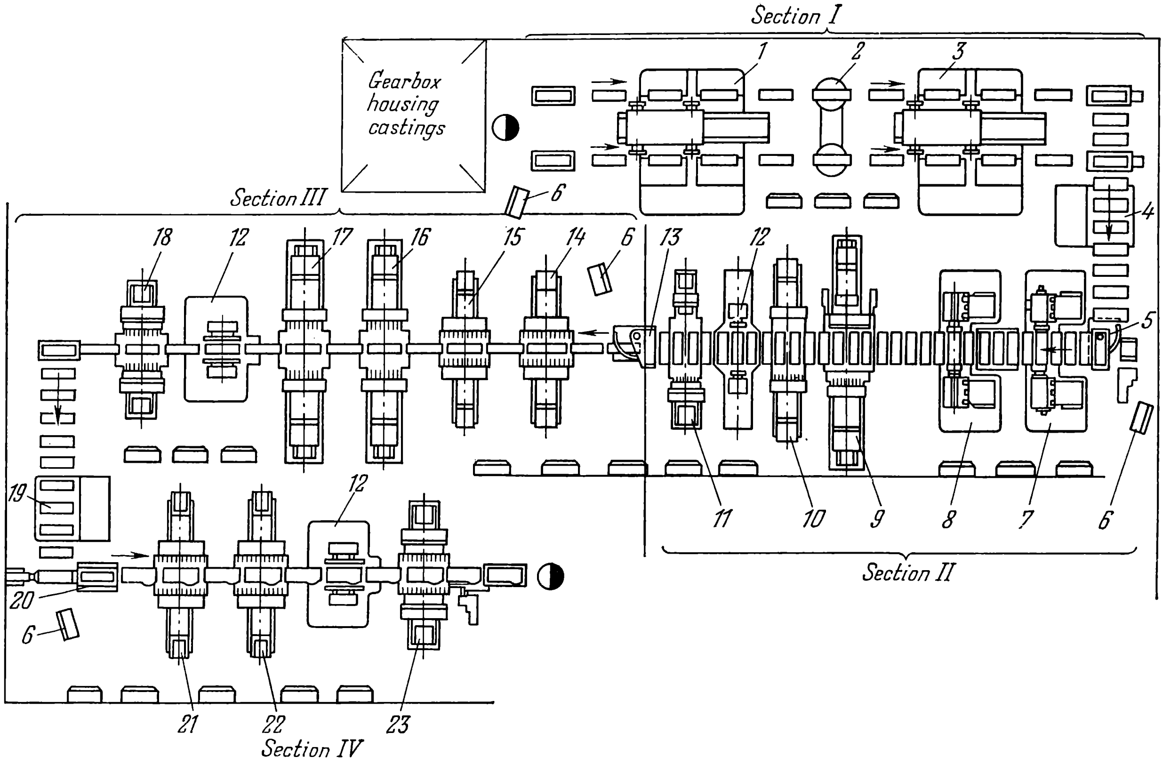


Fig. 218. Layout of a transfer machine for processing tractor gearbox housings:
1, *3*, *7* and *8*—milling machines; *2*, *5* and *13*—turntables; *4* and *19*—cross conveyers; *6*—section control desk; *9*, *10*, *14*, *15*, *16*, *17*, *21* and *22*—drilling, reaming and boring machines; *11*, *18* and *23*—tapping machines; *12*—fixture for checking the drilling depth; *20*—turning drum

and the table travel can be utilized for workpiece conveying without the need for a special conveyer drive (Fig. 220).

In this arrangement, two bars *1* with disappearing fingers *2* are secured on the machine table. Bars *3* with disappearing fingers *4* are secured alongside of bars *1* on the bed. During working travel of the table, workpiece *A* is milled while the other workpieces are transferred by fingers *2* along stationary bars *3*, sinking disappearing fingers *4*. In the rapid return stroke of the table, the workpieces are held from travelling back by fingers *4* while fingers *2* sink and pass under the workpieces. The clamping fixture returns with the table, releasing and leaving workpiece *A* and approaching workpiece *B* which it clamps.

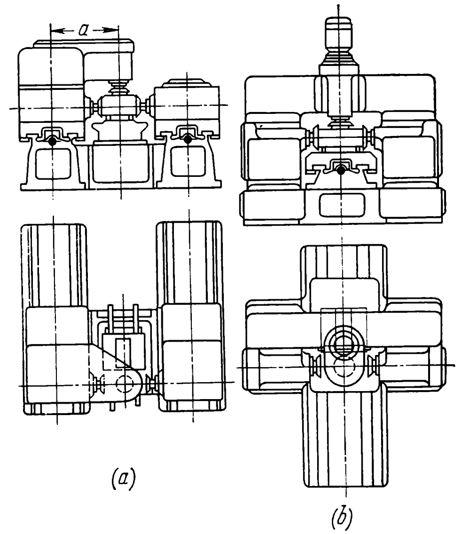


Fig. 219. Milling machines operating with (*a*) cutter feed and (*b*) workpiece feed

The shortcoming in machining by the method shown in Fig. 219*a* has been eliminated in milling automobile engine cylinder blocks at the Ford plant (USA). To simplify the feed drive of the milling heads and to raise its dependability (Fig. 222), milling heads *A* and *D*, and *B* and *C* are linked in pairs by tie-rods *E* and *F* and rack drives with shaft *I* and pinions *G*. The coupled milling heads are traversed by a common hydraulic drive, and the rack-and-pinion drives ensure synchronous operation, relieving the control system of this function.

The transfer bar is secured to head *A* and transfers the blocks along two hardened and ground bars during the rapid return stroke of milling head *A*. In this way, the conveyer drive is eliminated and the control system is simplified. This also increases the dependability of this section of the transfer machine and raises the utilization factor of its component equipment.

During the milling process, the blocks are held from above by hydraulic clamping facilities.

The block casting is loaded at station *1*. Single-spindle head *A* rough mills the upper jointing surface of the cylinder block (Fig. 221) at station *2* with a face-milling cutter having 38 inserted blades.

Five-spindle head *B* rough mills the mating surface for the crankcase at station *3* with two face-milling cutters, while at station *4*, with three side-milling cutters, it mills the recesses for the crankshaft main bearing caps and the mating surfaces for the caps. At stations *5* and *6*, two-spindle head *C*

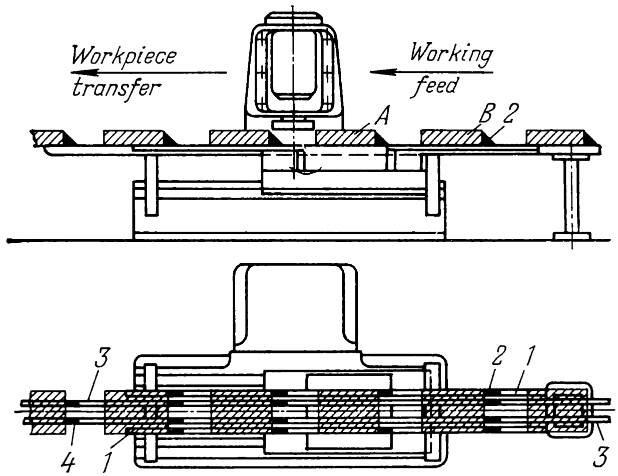


Fig. 220. Planer-type milling machine arranged along the axis of the transfer machine

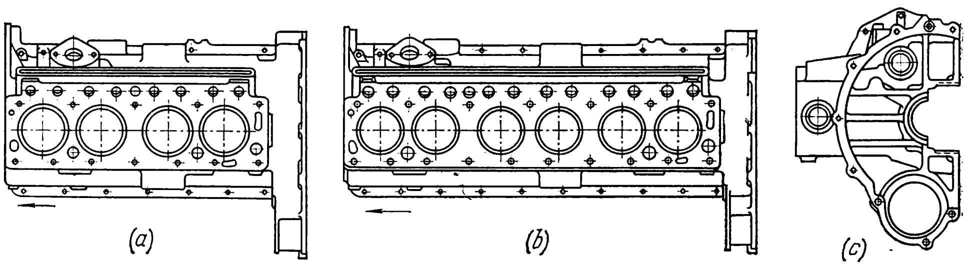


Fig. 221. Cylinder blocks of automobile engines:
(a) four-cylinder block; (b) six-cylinder block; (c) end view

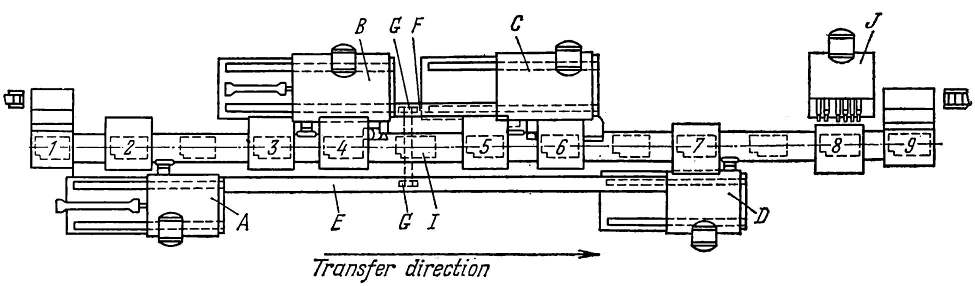


Fig. 222. Layout of the milling section

finish mills the mating surfaces for the crankcase and bearing caps, as well as the locking slot for the caps. Single-spindle head *D*, at station *7*, performs the semifinish milling operation on the mating surface for the cylinder head. Six-spindle power unit *J*, at station *8*, drills, enlarges and reams two locating holes from the crankcase side. At station *9* the engine block is turned with the crankcase side downward. In housing-type parts, holes are usually machined from several sides of the workpiece.

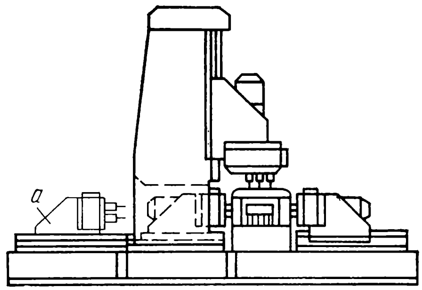


Fig. 223. Three-way multiple-spindle drilling machine with an arch-type column

If multiple-way machining is applied at each working station, the number of required machine tools is reduced. In a tunnel-type transfer machine (with in-line workpiece transfer through the clamping positions), three-way unit-built machine tools with three horizontal power units are inapplicable. If a machine tool with two horizontal and one vertical power unit is used, tool changing will require that one horizontal power unit pass through an opening in the arch-type column of the vertical power unit (Fig. 223). This lengthens the required travel for tool changing (position *a* in Fig. 223), leading to an increase in floor space occupied by the transfer machine and in the cost of the machine. For this reason, as a rule, only two-way drilling and boring machines are employed in tunnel-type transfer machines.

If the size of the workpiece is not large in the direction of its travel along the transfer machine, each power unit can machine two workpieces simultaneously at two adjacent stations, performing the same operation on the two workpieces in parallel or performing consecutive operations. This reduces the number of units of equipment and raises the dependability of the system.

Sometimes, due to the large expenditures on processing equipment, it proves economically expedient to apply three-way machine tools, making use of longitudinal and cross transfer, instead of the two-way type. The handling system, in this case, is somewhat more complicated, but the number of working stations and machine tools is reduced.

17-2. Transfer Systems of Transfer Machines for Housing-Type Parts

If housing-type parts have a sufficiently large bearing surface and can be guided at the sides with strips to prevent accidental swivelling, they can be conveyed by being pushed along bars or skids.

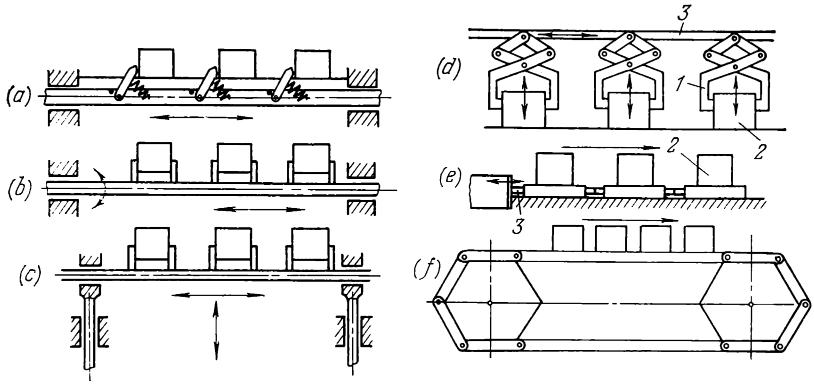


Fig. 224. Transfer bar and other periodic conveying systems

To avoid damage to the bearing surfaces of housings of aluminium alloys in sliding along the skids, they are raised by the handling device, transferred to the next station and lowered onto locating strips.

When parts are of such intricate shape that direct automatic location in machining and transfer is difficult and conveying along skids is impossible, they are clamped in individual fixtures or pallets which are transferred from station to station. These pallets are designed for convenient location of the workpiece in machining, transfer and fixing in the machine tools.

Transfer bars are used for moving housing-type parts through transfer machines. They simultaneously transfer all the workpieces in a section to the next stations. The pitch of the transfer bar is equal to the distance between the stations.

The following types of transfer bars find application: disappearing finger type (Fig. 224*a*); rotating finger type (Fig. 224*b*); walking beam type (Fig. 224*c*) and overhead type with grips (Fig. 224*d*). Periodic-action chain conveyers (Fig. 224*f*) are also used. The most popular of these are the disappearing finger type transfer bars (Fig. 225).

In this device, triangular index fingers *3* are mounted on pins secured in bar *1* which passes through the whole section of the transfer machine. Under the action of spring *2* the fingers project above the bar, bearing with their lower end against a pin in the bar. As the bar moves forward, the fingers engage the workpieces and push them forward one pitch. The bars are supported underneath by rollers (the rollers are in bracket *4* in Fig. 225; see also roller *1* for supporting bar *2* in Fig. 226). In transferring with a finger, overtravel of the workpiece may occur if it slides ahead away from the finger. It is necessary to limit the speed of such transfer bars to avoid overtravel.

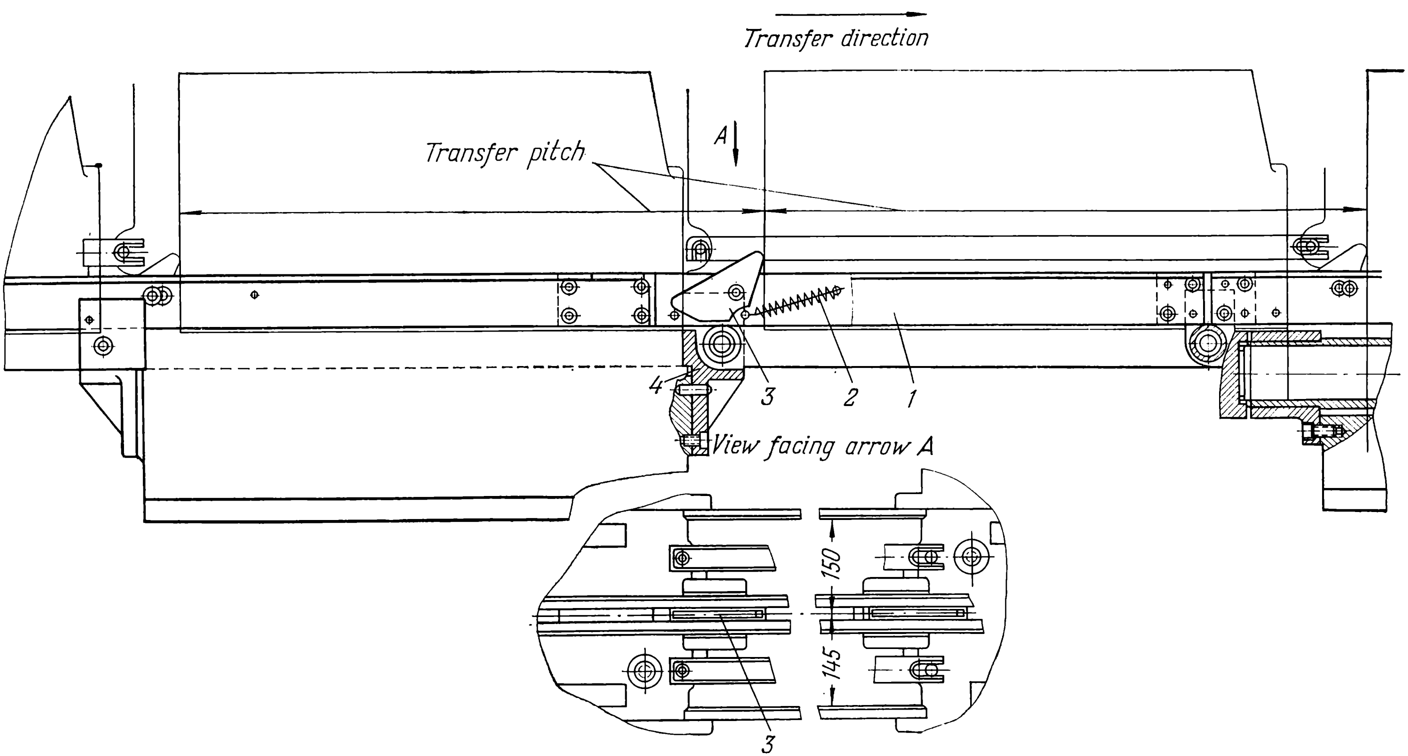


Fig. 225. Transfer bar with spring-loaded disappearing fingers

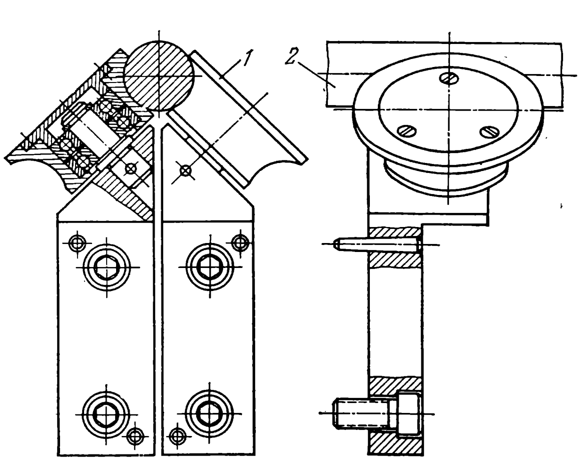


Fig. 226. Rotating finger-type transfer bar:
1—roller supports; *2*—bar

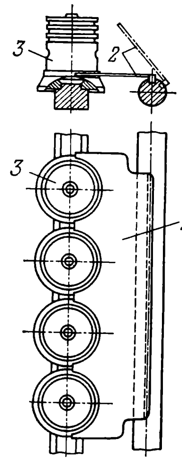


Fig. 227. Rotating finger-type transfer bar

This shortcoming is not found in the rotating finger transfer bar (Fig. 227) with a round transfer bar *1* carrying the four-piece rotating fingers *2*, between whose projections the workpieces *3* fit with some clearance. The size of this clearance determines the maximum overtravel. This feature enables the transfer speed to be increased. The design of the transfer bar is complicated by the necessity to rotate the bar to raise and lower the fingers in strict co-ordination with the operation of the workpiece locating devices.

Transfer bar systems have base and side guide strips for the workpieces (Fig. 228). The bars are usually actuated by a hydraulic cylinder on the following cycle: movement of the bar with the workpieces forward, slow approach of the workpieces to the working stations, operation of the piston rod of the hydraulic cylinder against a positive stop while the workpieces are being clamped in the working stations, rapid return of the bar to the initial position and a stop in this position.

This cycle has been applied in the hydraulic drive for a transfer bar with disappearing fingers designed by Special Design Office for Transfer Machines and Unit-Built Machine Tools (Fig. 229). In addition to the hydraulic cylinder, this drive includes a hydraulic control panel (type У2423), cam-operated flow-control valve (type СК46) and the mechanism for cam-operated unloading (type СК45).

This drive has the following operating cycle:

1. *Forward travel.* Solenoid *1* is energized. Reversing valve *7* is shifted to the upper position (in the diagram). Oil is admitted through the cam-

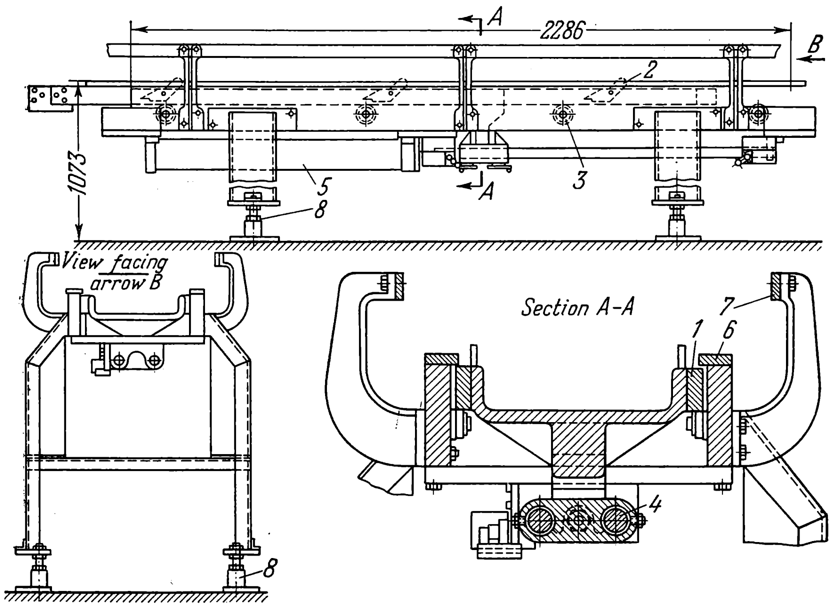


Fig. 228. Standardized disappearing finger-type transfer bar with a hydraulic drive: 1—bars; 2—fingers; 3—roller support of bar; 4—carriage guides of the transfer bar drive; 5—cylinder of the hydraulic drive; 6—base guide strips for the workpieces; 7—side guide strips; 8—jacks for height adjustment

operated flow-control valve to the head end of cylinder *11* and the bar travels forward.

2. *Slow approach* of the workpieces to the next working stations. The slowing-down cam *16* operates flow-control valve *15*.

3. *The piston rod bears against a positive stop*. Along a groove 1 mm wide on the plunger of the flow-control valve oil passes into the head end of cylinder *11*, forcing the piston rod against stop *13*. Pressure switch *12* energizes the solenoid for fixing and clamping the workpiece.

4. *Transfer bar return*. Solenoid *2* is energized and return travel begins, first at slow speed, until slowing-down cam *16* releases the pin of the cam-operated flow-control valve, and then at high speed (if there is a check valve in the cam-operated flow-control valve, as in Fig. 232, rapid return begins immediately).

5. As the *bar approaches the initial position*, unloading cam *17* releases the latch of unloading valve *18*, the spring shifts the valve spool upward (in the diagram), and both pumps, *19* and *20*, unload to the tank through the valve

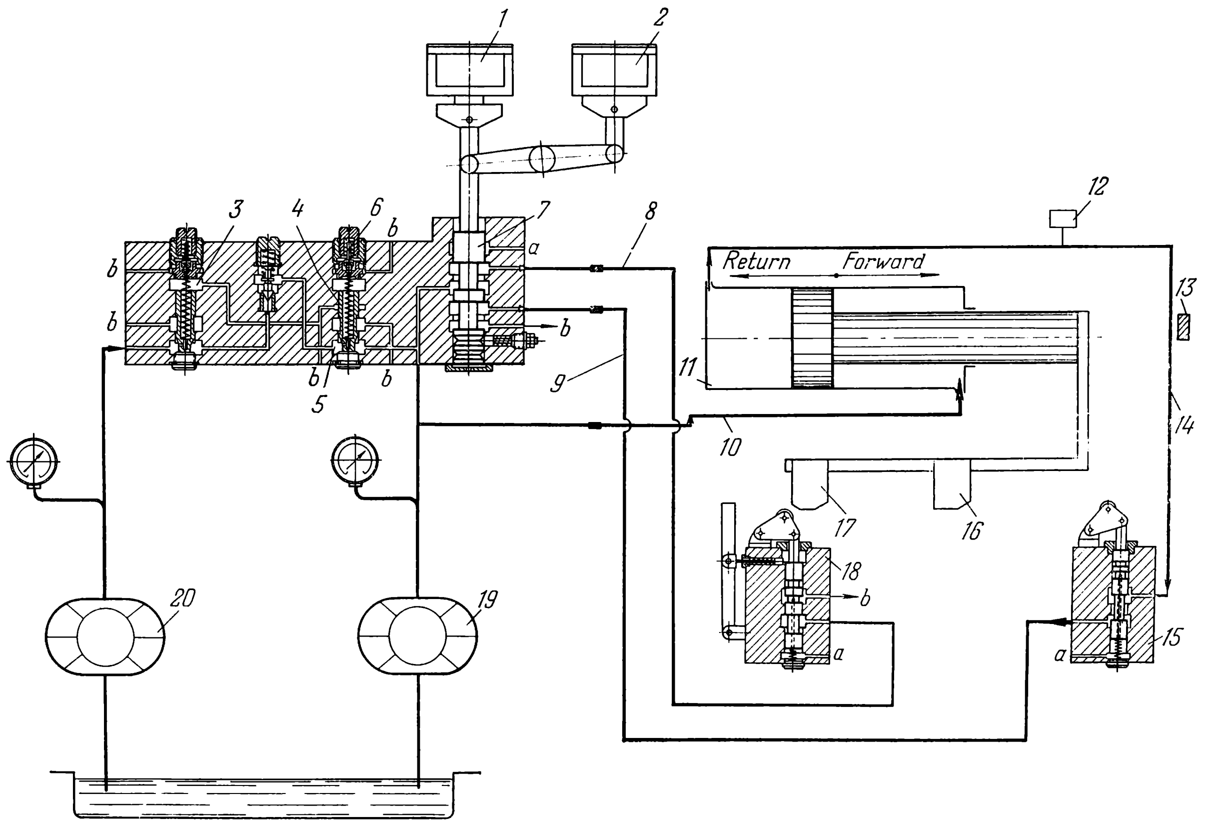


Fig. 229. Hydraulic circuit diagram of a transfer bar drive:

1—solenoid for forward travel; *2*—solenoid for return travel; *3* and *4*—chambers preceding the relief valves of pump *20*; *5*—groove in the valve body; *6*—relief valve of pump *19*; *7*—transfer bar reversing valve; *8*—pipeline from hydraulic control panel to the unloading valve *18*; *9*—pipeline from the control panel to the flow-control valve; *10*—pipeline from the rod end of the hydraulic cylinder to the control panel; *11*—hydraulic cylinder of the transfer bar drive; *12*—pressure switch; *13*—transfer bar stop; *14*—pipeline from the head end of cylinder *11* to flow-control valve *15*; *15*—cam-operated flow-control valve; *16*—slowing-down cam; *17*—unloading cam; *18*—cam-operated unloading valve; *19*—high-pressure pump; *20*—low-pressure pump

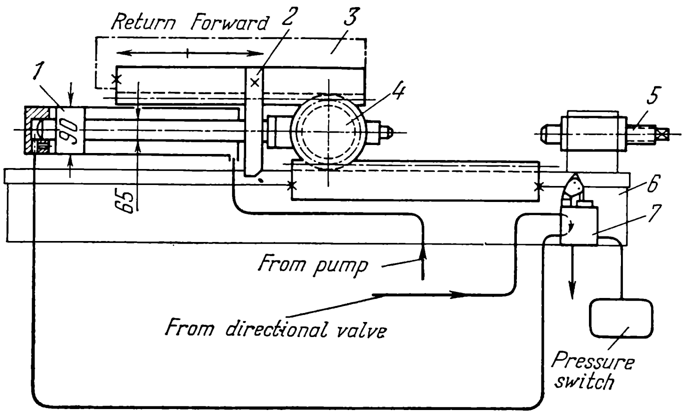


Fig. 230. Unified transfer bar drive

The combined hydraulic circuit and kinematic diagram of a unified drive for a transfer bar (developed in the Special Design Office) is shown in Fig. 230. Through the speed-up rack-and-pinion drive *4*, the differential cylinder, mounted on bed *6*, traverses table *3* on which is mounted a bracket linked to the transfer bar member. The transfer bar is slowed down before reaching stop *5* by means of cam-operated flow-control valve *7* which is actuated by cam *2*. This reduces the amount of oil admitted to the head end of cylinder *1*.

As the transfer bar slows down, workpiece *1* (Fig. 231) will not slide ahead, away from finger *2*, if its inertia force R_a is equal to or less than the friction force R_{fr} of the workpiece on strips (skids) *3*, i.e., if

$$ma \leqslant Gf \tag{80}$$

where m = mass of the workpiece
a = its deceleration
$G = mg$ = weight of the workpiece
f = coefficient of friction of the workpiece sliding along the guide strips (skids).

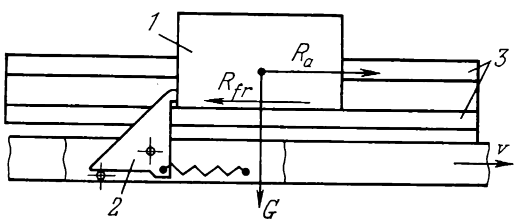


Fig. 231. Diagram of forces acting on the workpiece when the transfer bar is slowed down

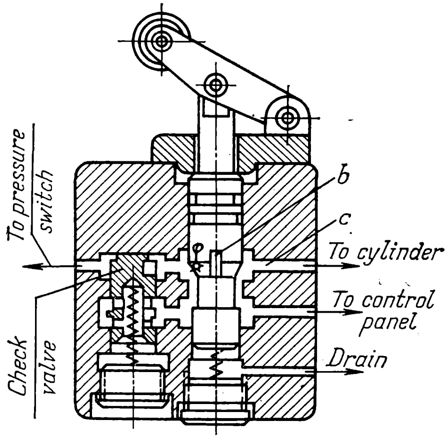


Fig. 232. Cam-operated flow-control valve with a check valve

From this the permissible deceleration is determined

$$a \leqslant gf = 9.81\,f \text{ m per sec}^2 \quad (81)$$

Therefore, at $f = 0.05$ to 0.1, deceleration a should not exceed 0.5 to 1 m per sq sec. At a constant deceleration a and a workpiece transfer speed of $v = 14$ m per min, the length of slowing-down travel is within the limits of about 25 to 45 mm. The spool of the cam-operated flow-control valve (Fig. 232), having a tapered throttling surface with a taper angle of $\varphi = 6°$, does not provide constant deceleration, and the length of travel in slowing down the bar ranges from 30 to 60 mm at $v = 14$ m per min. To make sure that the transfer bar is advanced tightly against the stop, a groove 1 mm wide is provided on the spool to admit a small amount of oil into the cylinder when the spool completely closes off port *c*. It follows from equation (81) that the magnitude of deceleration a, and consequently the length of travel, do not depend upon the weight of the workpiece but only on its speed and the coefficient of friction f. The length of deceleration is set up for various speeds by selecting a suitable angle of inclination for cam *2* (see Fig. 230). When the transfer bar runs up against the stop, the pressure in the head end of the cylinder increases. This actuates the pressure switch, transmitting a signal for energizing the solenoids of the hydraulic control panel for clamping the workpiece.

At the beginning of the return travel of the transfer bar, the oil from the head end of the cylinder drains through the check valve back to the tank.

As the transfer bar approaches its initial position it is braked before stopping by a cushioning arrangement built into cylinder head *1* (Fig. 230). The piston is braked over a length from 5 to 15 mm.

Only cam-operated flow-control valves are used in transfer bar drive hydraulic circuits to slow down the bar as it approaches the positive stop, since it is necessary to actuate the pressure switch, installed after the flow-control valve, only when the transfer bar is tight up against the stop.

If a cushioning device, built into the cylinder, were used for this purpose, the final forward position of the transfer bar would be indefinite since the pressure switch could be actuated during the cushioning motion. In addition, the use of a flow-control valve allows the length of slowing-down travel to be adjusted by changing the profile of the cam.

The type of transfer bar drive shown in Fig. 230 is applied for bar strokes from 400 to 1,100 mm at workpiece speeds from about 8 to 22 m per min.

Transfer bars with a mechanical drive are not frequently used. They are operated by either a link drive or a chain, both ends of which are attached to the bar. The driving sprocket of the chain is powered by an electric motor through a reversible transmission with friction clutches.

Walking beam transfer bars (see Fig. 224*c*) are seldom employed, and then only when housing-type parts cannot be pushed along skids, for example, to avoid damaging the base surface in machining aluminium housings, or when the parts are unstable and must be clamped during the transfer movement.

Overhead transfer bars with automatic grips (see Fig. 224*d*) are used chiefly in machining shafts.

Pusher-type transfer conveyers (see Fig. 224*e*) are extremely simple in construction. The pusher, usually the rod of a hydraulic or pneumatic cylinder, pushes the whole line of workpieces one pitch. If the workpieces are heavy, a supplementary pusher with a short stroke may be used to start the workpieces.

Because of the accumulated error the workpieces must be located, with the pusher retracted, consecutively, beginning with the workpiece farthest away from the pusher. This lengthens the cycle time and is a drawback of a pusher-type conveyer, limiting its field of application to the return of fixtures or pallets (see p. 321). But in this case, an excessively large amount of pallets is required to form a continuous line, in comparison with the amount required for operation, this again being a shortcoming of this system.

Chain-type transfer systems (Fig. 224*f*) do not provide workpiece transfer sufficiently accurate for fixing the workpiece in its location and clamping.

17-3. Turning Devices

Turning devices are employed in transfer machines for changing the orientation of housing-type parts in certain sections of the machine.

Depending upon the manufacturing process being applied, the following types of turning devices are used: drums for turning a workpiece about a horizontal axis (Fig. 233*a*), tables for swivelling a workpiece about a vertical axis (Fig. 233*b*) and turnover devices for turning workpieces about an inclined axis (Fig. 233*c*).

To avoid lengthening the cycle time and reducing the output of the transfer machine, it is desirable that the construction of the turning device and its drive be such as to enable the time required for turning to be overlapped by the machining time. It is also desirable that the turning device be controlled from the hydraulic panels already incorporated in the transfer machine.

A standard turntable (Fig. 234) allows swivels of either 90° or 180° to be performed by adjusting the length of the rod stroke with stops.

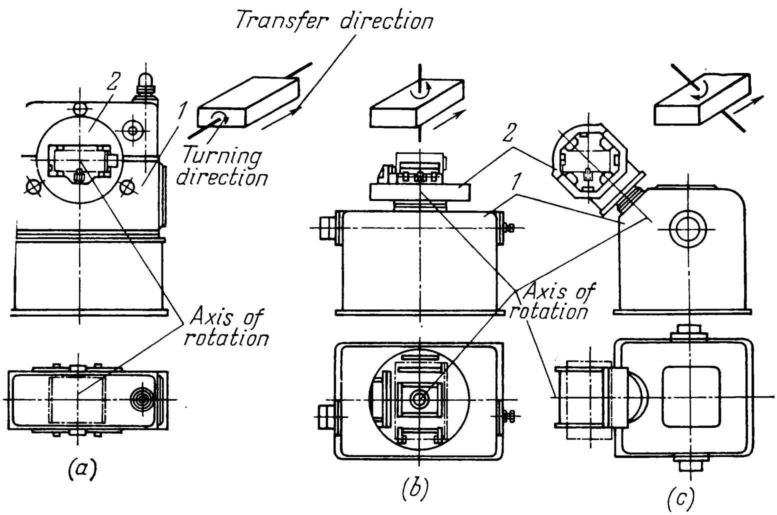


Fig. 233. Types of turning devices:
1—base; *2*—turning member

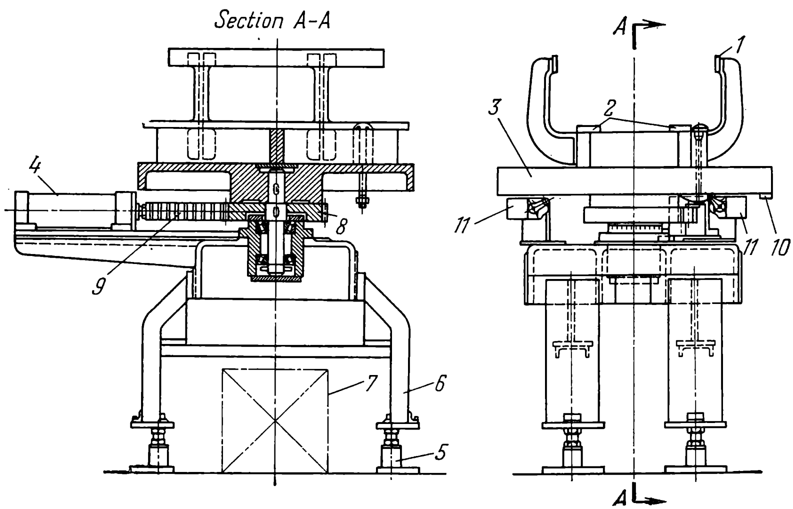


Fig. 234. Standard turntable for swivelling workpieces 180° in a horizontal plane:
1—side guide strips of the turntable; *2*—base guide strips; *3*—turntable with the rack pinion *8*; *4*—cylinder with the rack teeth cut on piston rod *9*; *5*—jacks for height adjustment of the turntable; *6*—arch-type turntable base; *7*—chip conveyer; *10*—control cam dog; *11*—limit switch

Turntables are employed when the sections preceding and following the turning operation have independent transfer bars.

Overhead socket-wrench-type turning devices are used when the sections of the transfer machine preceding and following the turning station have a common transfer system.

The unit shown in Fig. 235 turns pallets with clamped workpieces (globe valve bodies) 90° at the end of section *1* of the transfer machine (see Fig. 239). As socket *2* descends, its rectangular lugs enter recesses in the baseplate of the pallet (fixture). The socket is mounted on a spline shaft attached to the rod of the socket-lifting cylinder *4*. The socket is turned together with the pallet by means of a rack-and-pinion drive actuated by hydraulic cylinder *3*. After turning the pallet, the socket is lifted and turned back to the initial position. The turned pallet is pushed by the transfer system to the next station while the next pallet is transferred to the table of the turning station. Crossed slots underneath the pallet are engaged by the guide rail of the transfer system. The phases of the cycle are controlled by cams fastened to the lugs on the rods of hydraulic cylinders *3* and *4*.

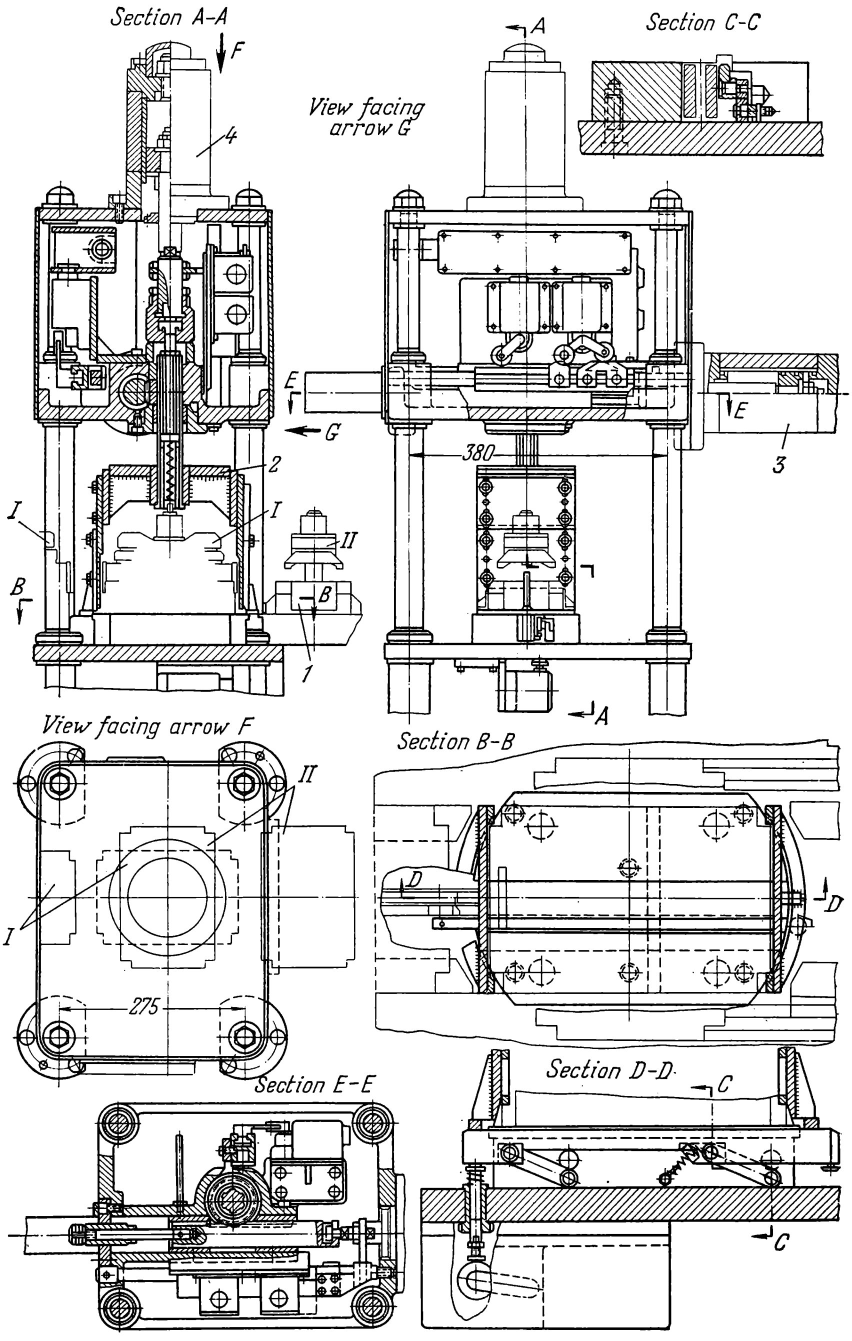
The turntable (Fig. 236) serves for lowering the pallet (fixture) to the level of the transfer system, returning it and turning it back to the initial position at the loading station. The baseplate of the pallet, carrying the workpiece, is transferred to the table which is mounted on a bar with a bayonet slot. The bar is fastened in a bridge-type member which is linked to the rod of a hydraulic cylinder and also carries two guide bars and a bracket with cams for controlling the phases of the cycle.

The turning devices shown in Figs. 234 and 236 carry out workpiece swivel in six operation elements (taking into consideration transfer bar movement). The turning member is swivelled twice, once with and once without the workpiece. The time required for swivelling is not completely overlapped by any element of the machining cycle in this section of the transfer machine.

A drum for turning the workpiece 90° about a horizontal axis (Fig. 237) consists of base *1* and the turning member on which the workpiece is placed. The turning motion is actuated by hydraulic cylinder *12* through piston rod *11* with integral rack teeth, pinion *10*, ratchet mechanism *13*, gear *9* and gear rim *7* mounted on turning drum *6*. During the return stroke of the piston rod, drum *6* remains stationary due to the action of locking member *4* which enters a seat in one of the four blocks *5*, press-fitted into the turning member. Disks *3*, screwed to the turning member, rest on rollers *2*. On one side of the turning member the rollers have a bead fitting into a groove of disk *3*. This locates the drum in the axial direction. The extreme position of the rack is controlled by stop *8*.

The turning drum of this device operates in three operation elements:

1. In its forward travel, the transfer bar of the first section enters the next workpiece into the drum, pushing out the previously turned workpiece to an


Section A-A
F
4
View facing arrow G
A
Section C-C
E
G
E
380
3
2
I
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Section D-D
C
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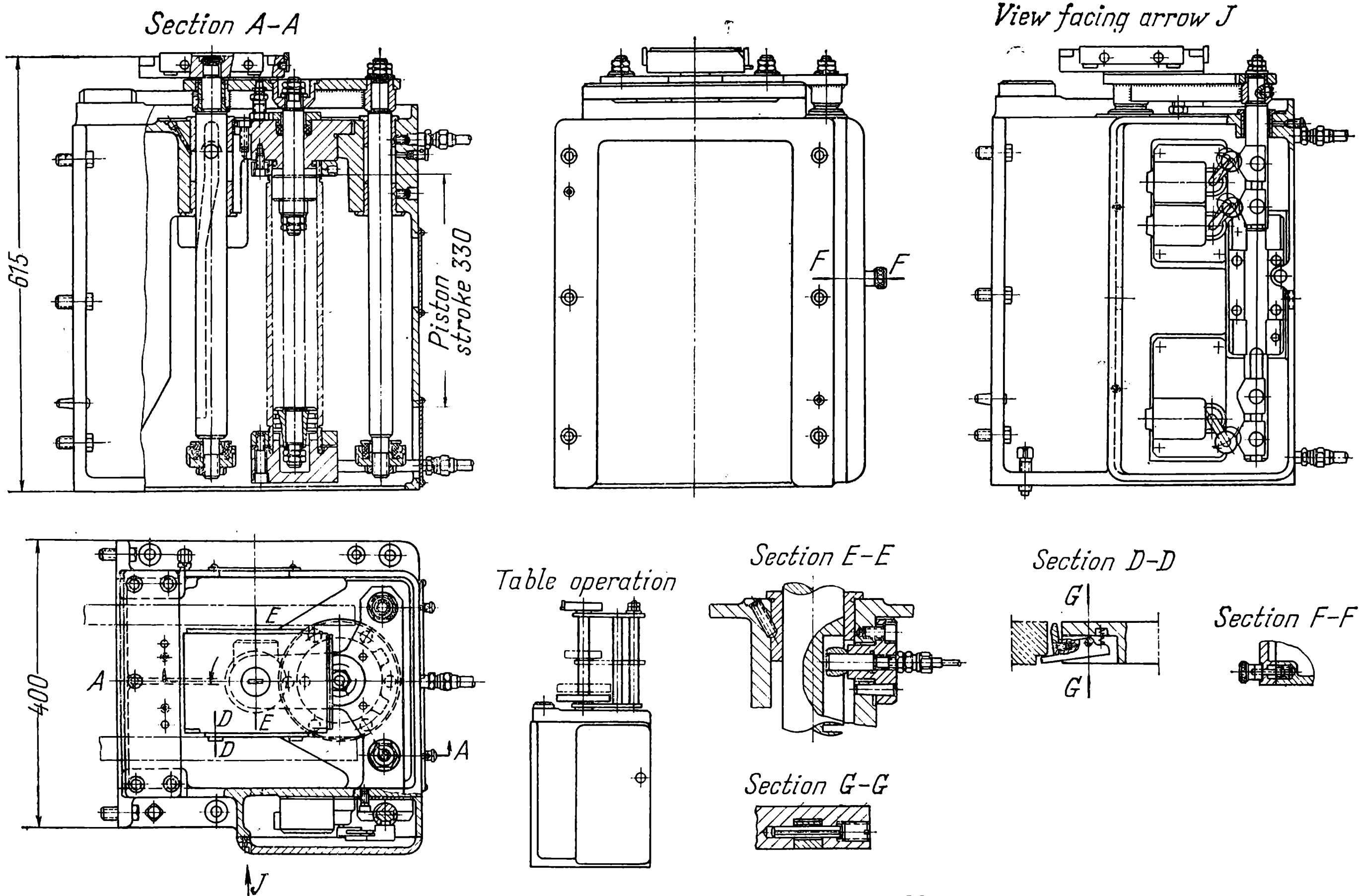


Fig. 236. Lifting-type turntable

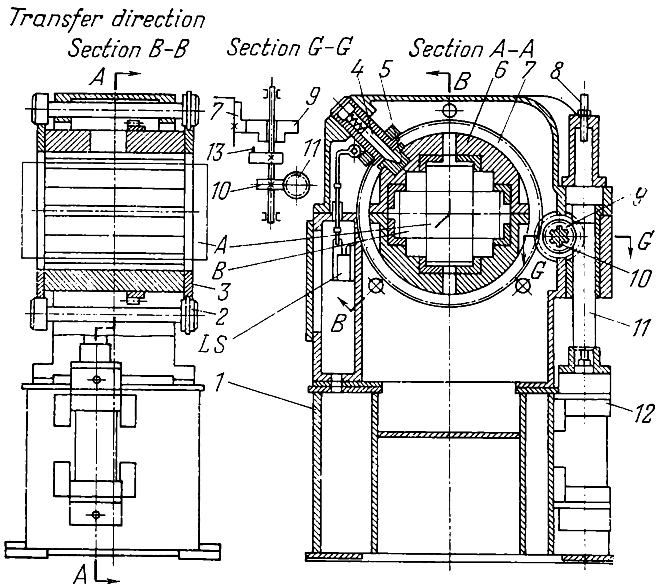


Fig. 237. Drum for turning a workpiece 90° about a horizontal axis

intermediate station outside of the drum. At this moment, the transfer bar of the second section is in the FORWARD position.

2. Both transfer bars are retracted.

3. The drum is turned simultaneously with forward travel of the transfer bar of the second section in which the workpiece is moved from the intermediate station.

There is no need for returning the turning member to any initial position since, in each of its four indexed positions, the drum can accept the next

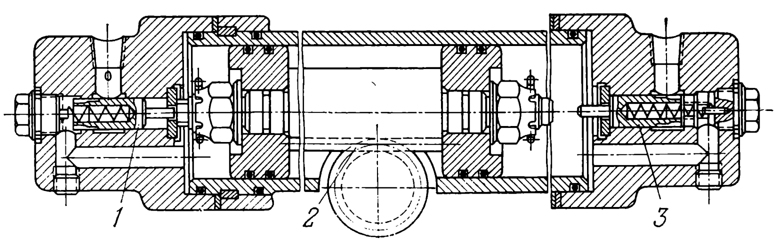


Fig. 238. Hydraulic cylinder with a rack-and-pinion drive

workpiece in the position it has in the first section and deliver a turned workpiece simultaneously to the intermediate station.

The motion of the rod of the hydraulic cylinder forward and back can be made to coincide with the clamping and unclamping of the workpiece, and the whole turning cycle is overlapped by the machining cycle. Such designs are called through-type turning devices. They are more advanced than other types since they do not increase the cycle time and allow the turning and clamping controls to be combined.

Illustrated in Fig. 238 is the longitudinal section of a hydraulic cylinder with a rack-and-pinion drive *2* used for turning a table. The table is braked in the two extreme positions by a damping arrangement consisting of two throttle valves *1* and *3*, inserted in the cylinder heads.

Transfer Systems in Machining Workpieces in Pallets

Parts of intricate configuration, with no surfaces that are convenient for dependable location in transfer and machining, are clamped in individual fixtures when they enter the transfer machine and are transferred with the fixtures from station to station. These fixtures, maintaining the orientation of the workpiece throughout the machine, are located and clamped at the working stations. They are frequently called *pallets* and the machines are called *palletized transfer machines*. The rear axle housings of automobiles, oil pump bodies and other parts of complex shape are machined with pallets. The application of pallets extended the type range of parts that can be transfer machined since the intricate shape is no longer a decisive factor in determining whether transfer machining is feasible.

In transfer machines that are changed over for handling a group or family of parts, palletizing is used to unify the location, fixing and clamping of the various parts in the group at the different working stations. In this case, the part may be clamped to an intermediate individual adapter (plate or angle) which is located in the general pallet on unified datum surfaces and clamped (see Section 17-5).

At the end of the transfer machine, after the workpiece has been machined, it is removed from the pallet, and the pallet is returned to the loading station. This requires an additional conveyer which complicates the handling system in palletized transfer machines. Frequently, the workpiece is not removed from the pallet after machining and is returned with the pallet by a conveyer to the initial position—the unloading station—arranged adjacent to the loading position and tended by the same operator.

Depending upon the location of the conveyer for returning the pallets to the loading station, distinction is made between several types of palletized transfer machine layouts.

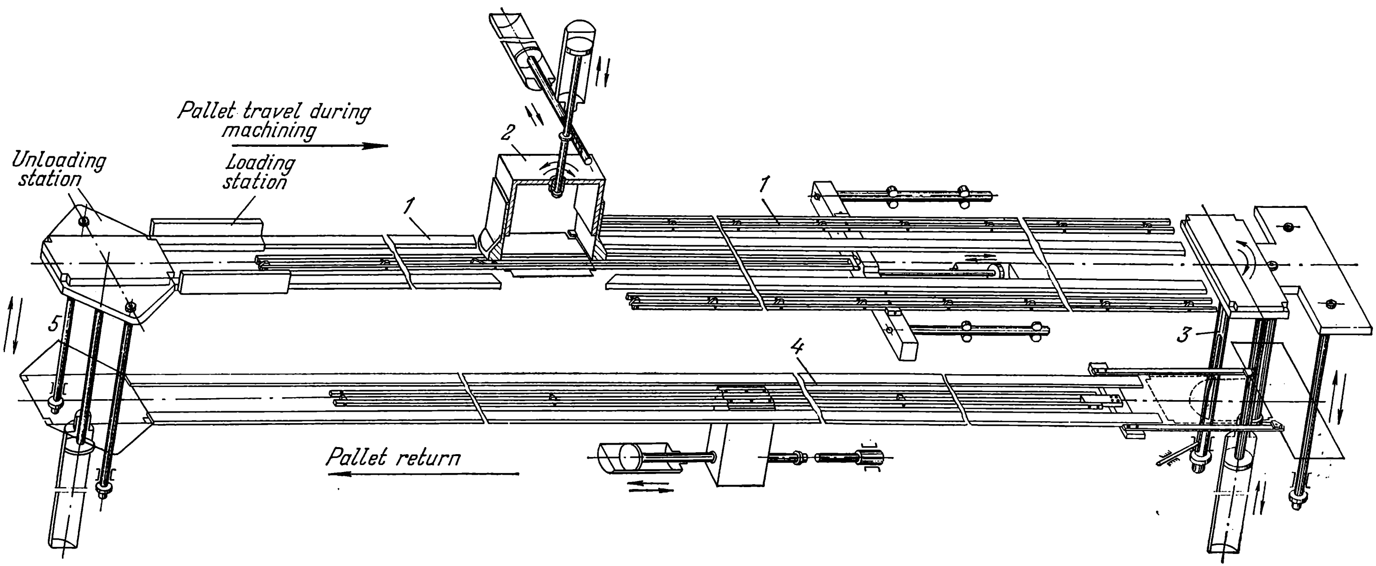


Fig. 239. Handling system of a palletized transfer machine for machining globe valve bodies

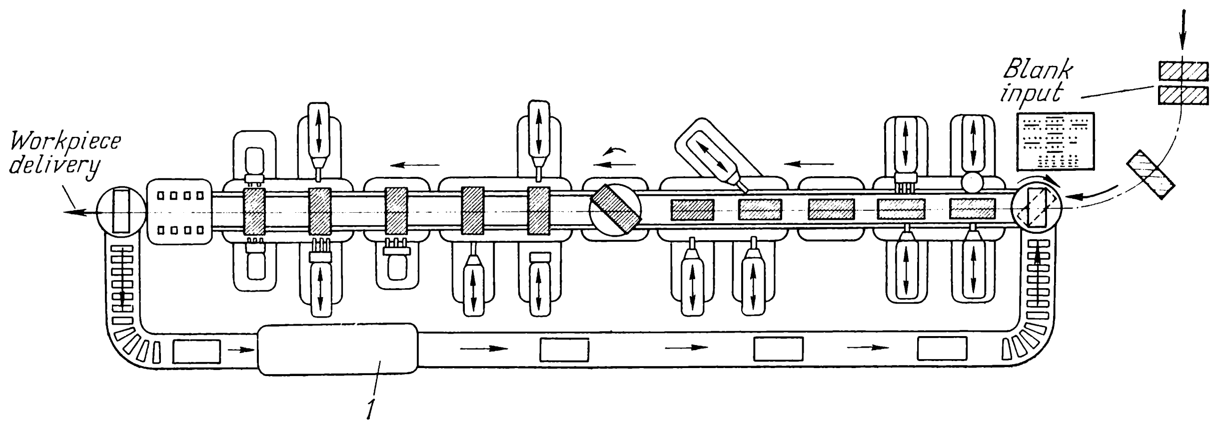


Fig. 240. Layout of a transfer machine with frontal arrangement of the pallet return conveyer

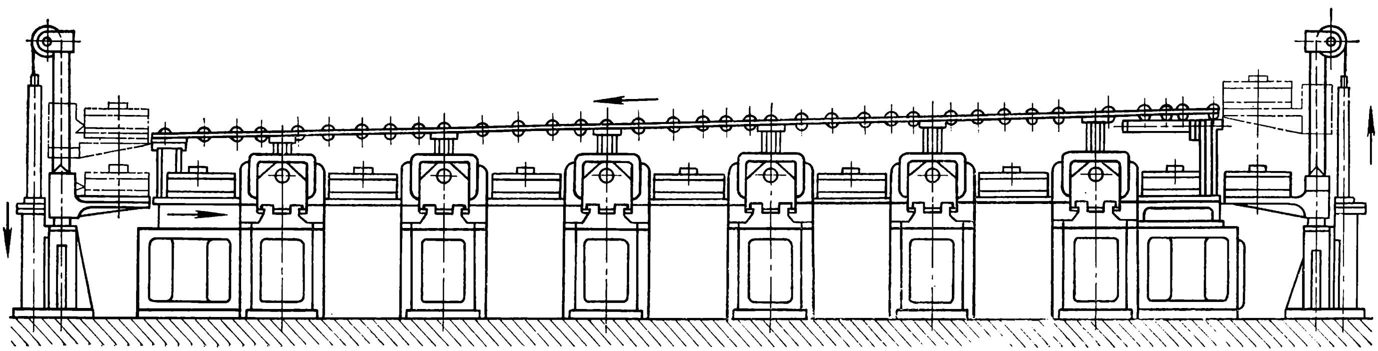


Fig. 241. Pallet return on an overhead inclined roll table

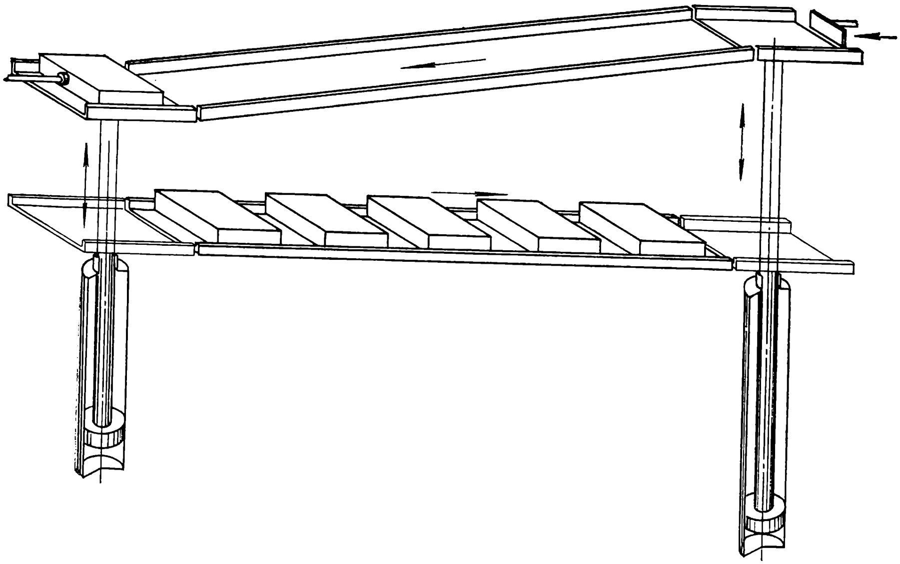


Fig. 242. Pallet return system with an overhead inclined chute

In the arrangement shown schematically in Fig. 239, the transfer bar return conveyer *4* has a higher transfer speed and a longer pitch than the main transfer bar *1* under which it is located. In the process of simultaneous machining, two globe valve bodies clamped on a single pallet are turned 90° together with the pallet by the socket-wrench-type turning device *2* (see Fig. 235). Therefore, on being lowered to the level of the return conveyer, the pallet is turned back to the initial position by lifting turntable *3* whose construction is shown in more detail in Fig. 236. From the return conveyer. the pallet together with the finished valve bodies is lifted by lift table *5* to the unloading station where the workpieces are unclamped and removed.

This arrangement does not require additional floor space for returning the pallets but presents inconveniences in chip disposal and in maintenance of the return conveyer.

In a frontal arrangement of the return conveyer (Fig. 240) parallel to the line of machine tools, the equipment of the transfer machine is more accessible for servicing, except for the machine tools within the loop formed by the conveyers, but the occupied floor space is considerably increased in comparison with an arrangement having the return conveyer underneath the main transfer line. In return conveying (usually by means of a chain conveyer), the pallets pass through washing station *1* for cleaning chips

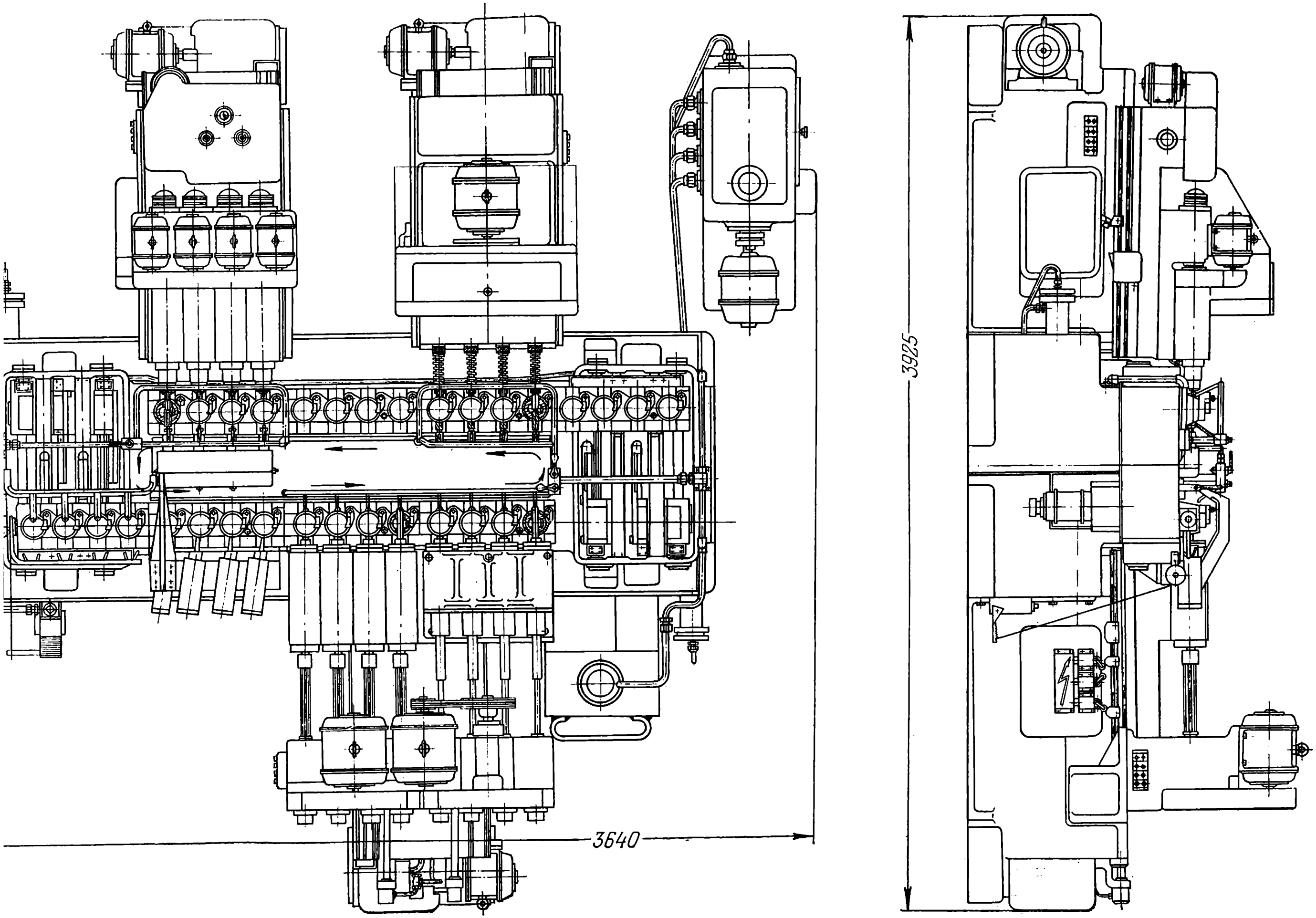


Fig. 243. Transfer machine for machining the wristpin bore in pistons clamped in sets of four on pallets

and metallic dust from the locating surfaces. This type of layout is the one most widely applied in palletized transfer machine design.

When there are no vertical machine tools in the transfer machine, the pallets can be returned by a conveyer arranged above the main transfer system and the machining zones. The return conveyer can be simplified by designing it as an inclined roll table (Fig. 241) or, if the transfer machine is short, in the form of an inclined chute (Fig. 242).

To avoid impacts at excessive speeds of pallet return in such conveyers, it is necessary to regulate the slope of the conveyer or to provide braking devices. In this arrangement all the machine tools are accessible, as are their working zones, and no additional floor space is required, but all the component machine tools must have a horizontal layout.

The need for a return conveyer is excluded if it is replaced by a section of the transfer machine with reverse transfer of the workpieces from station to station toward the initial loading station (Fig. 243).

At the ends of the sections, the workpiece with the pallet is transferred from one section to the other by a turning device (if the workpiece is to be machined only from one side), by a pusher (if the workpiece is to be machined on the other side in the return section), or by a conveyer (if the workpiece is machined on both sides by two-way machine tools in both the forward and return sections of the transfer machine). The main shortcoming of this layout is the poor accessibility to the working zones and to the machine tools within the loop formed by the transfer system.

17-4. Mechanisms for Locating and Clamping Housing-Type Parts

When the transfer system delivers a workpiece into a fixture, the workpiece is tentatively oriented by the locating elements of the fixture. Final location of the workpiece is to a previously machined surface (in most cases) which usually serves as the surface along which the workpiece slides in transfer, and two previously machined accurate holes in this surface. At the station, locating pins enter these two holes. After this, the workpiece is secured by special clamping devices of the special fixtures mounted at the stations.

The various versions of datum surface selection, depending upon the shape and arrangement of the machined surfaces, are shown in Fig. 244.

In order to avoid a loss of accuracy in setting up the workpiece, the locating devices should not be subjected to the clamping and cutting forces. The clamping facilities are usually actuated by hydraulic cylinders, either directly or through self-braking mechanisms (to obtain large clamping forces). In the last case, the clamping and locating mechanisms are actuated by a single hydraulic cylinder (Fig. 245).

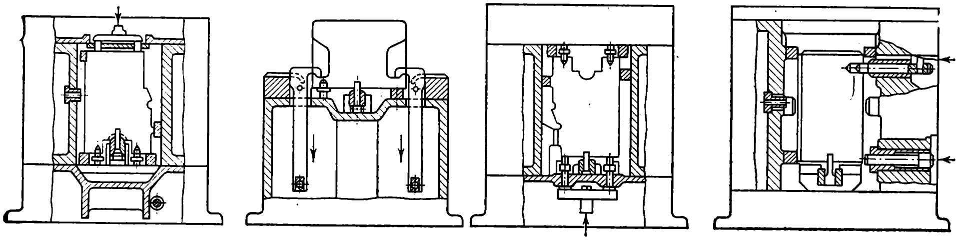


Fig. 244. Versions of datum surface selection

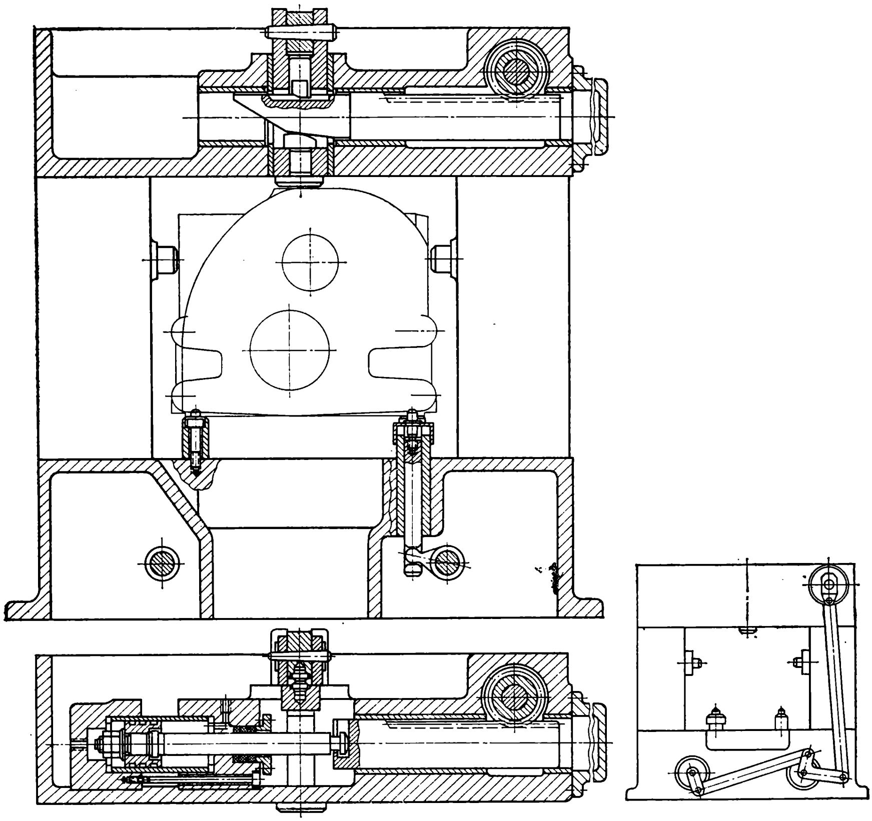


Fig. 245. Fixture for locating and clamping a gearbox housing by means of a flat wedge

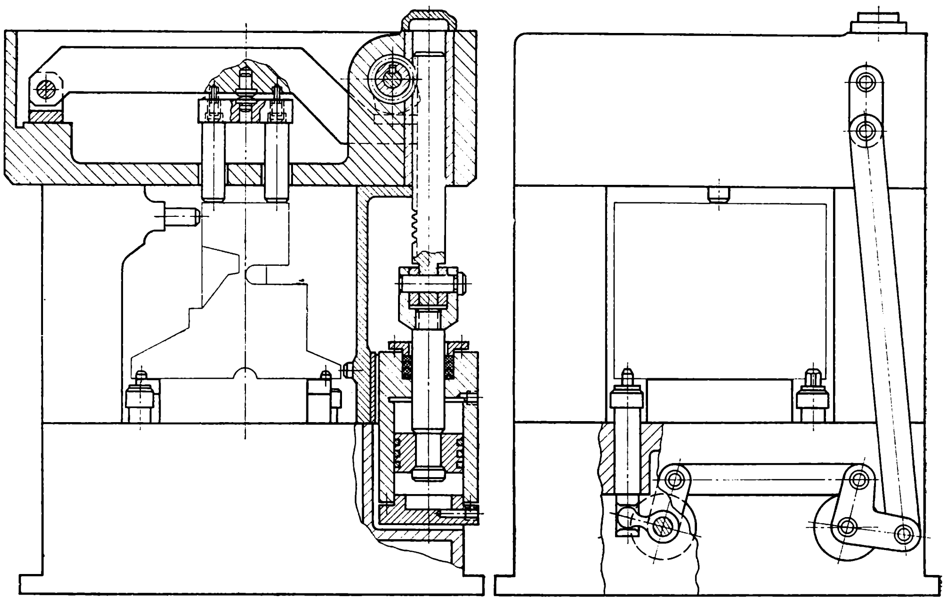


Fig. 246. Fixture for locating and clamping a cylinder block by means of a cam

In self-braking clamping mechanisms, the unclamping force is 10 to 20 per cent larger than the clamping force. Therefore, in clamping, the oil is delivered to the rod end of the cylinder, or an additional relief valve set at a lower pressure is installed in the line connecting the directional valve to the clamping end of the cylinder.

In clamping devices with direct clamping and with a separate hydraulic drive for actuating the locating devices it is more reliable to operate the clamping cylinders, at the end of the locating operation, by means of limit switches controlling the completion of locating pin insertion into the datum holes, than by the transmission of a hydraulic signal when the pressure increases in the locating pin cylinders. In the latter case, the command signal for clamping may be transmitted prematurely due to jamming of the locating pins by chips or against the workpiece if the transfer bar does not stop in the correct position.

Proper operation of the clamping facilities is checked by hydraulic pressure switches. In devices with a wedge-type clamping element, unclamping can be more reliably checked by an electrical limit switch because the pressure switch may be operated if the wedge jams. Location and clamping of the workpiece require from 0.03 to 0.1 min.

The specified magnitude of the clamping force is maintained constant by the hydraulic or pneumatic circuit during the whole time the workpiece is being machined. Special interlocking prevents the clamping force from being increased.

A fixture in which the workpiece is clamped by means of a cam is illustrated in Fig. 246.

Openings with inclined walls are provided in the lower plate of the fixture (see Fig. 245) to allow the chips to fall through freely.

17-5. Change-over Transfer Machines for Housing-Type Parts

Of all the types of transfer machines, the most difficult to change over for another part are the ones for housing-type parts. Various methods are applied to change over such transfer machines to handle a "family" of parts.

Change-over for machining different parts of the same type can be accomplished by designing the parts so that they can be handled in one transfer machine.

Thus, for example, with this in mind, the blocks of four- and six-cylinder engines (see Fig. 221) are designed with identical principal dimensions, such as: cylinder bore diameter, distance between the cylinder axes, distance from the datum end surface, block height, etc.

Additional lugs are provided on the castings for location in the fixtures. All holes in the two blocks are made identical in size and position. If one hole is absent in one block, a recess is provided to avoid removing a drill required for machining the other block. All machining that differs in the two engine blocks is transferred to the last transfer machine in the production line to simplify change-over of the main transfer machines.

In transfer machines made up of unit-built machine tools, change-over can be accomplished by changing the multiple-spindle heads and jig plates at the power units and by adding intermediate individual fixtures for holding the workpiece in unified pallets. In this case, location, fixing and clamping of the pallet remain the same when the transfer machine is changed over and a different workpiece is held in a different intermediate fixture.

One example of this principle is the Sundstrand six-station transfer machine (USA) for drilling, enlarging, boring and reaming holes in five different housing-type parts of hydraulic units (Fig. 247). Six tables carry two power units each with multiple-spindle heads for two adjacent machining stations. Working feed and rapid traverse motions of the tables are accomplished by means of feed gearboxes powered by electric motors rated at about 0.75 kW.

Graduated scales facilitate the setting of the command cams (stops) controlling table motions in changing over the machine

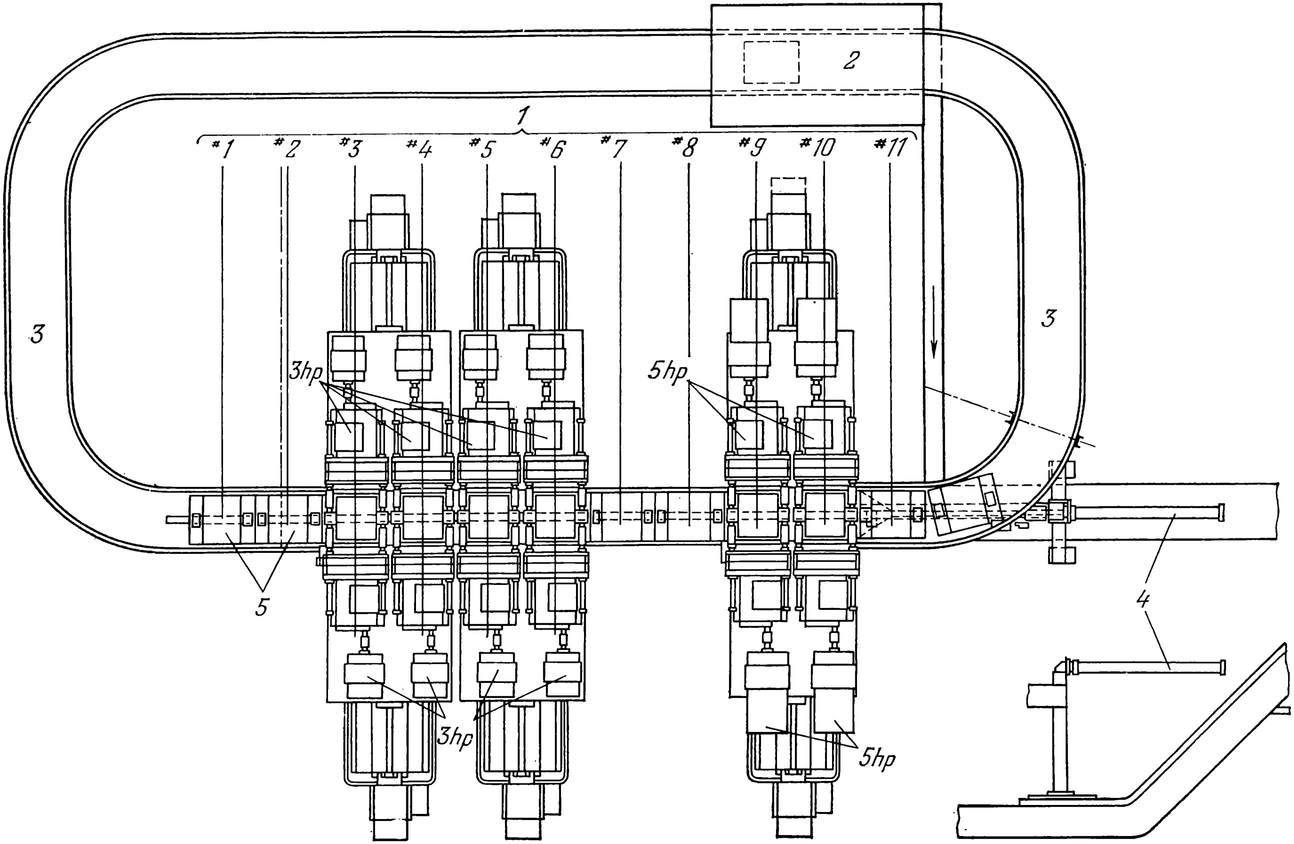


Fig. 247. Six-station change-over transfer machine made by the Sundstrand Machine Tool Co. (USA) for machining housings of hydraulic units:

1—arrangement of the working stations; *2*—washing machine; *3*—roller conveyer for returning the pallets with workpieces; *4*—pneumatic cylinder of the transfer bar system with a stroke of 19 or 38 inches for machining one or two workpieces at a time; *5*—unloading and loading stations

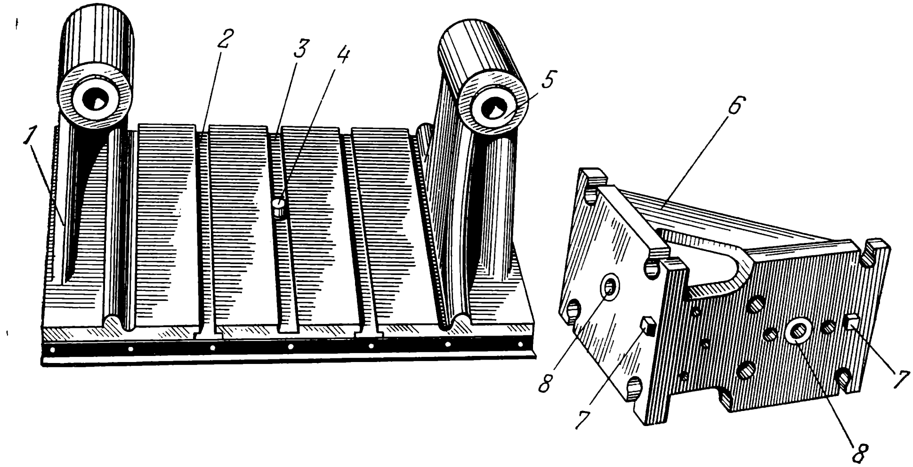


Fig. 248. Pallet (fixture) with an interchangeable holder for one of a group of different workpieces

The multiple-spindle heads are secured to the power units by four bolts. The heads can be quickly replaced in change-overs and are stored, together with the spring-loaded jig plates, on special racks having slide-out shelves on rollers and hoists suspended on monorails.

Interchangeable holders *6*, specially designed for each different part, are mounted together with the clamped workpieces on the pallets (Fig. 248), one set (12 pcs) of pallets being used for machining all five different housing-type parts.

Each pallet has a hardened and ground steel lower datum plate with two hardened and precisely ground bushings for the locating pins.

On the top surface of the pallet there are two T-slots *2* for bolts, a central keyway *3* and a central pin *4*, accurately located in respect to guide bushings *5* in the heads of two uprights *1* of the pallet. The pallet slides along the guide strips of the transfer system. It is located automatically by two pins entering the bushings in the baseplate, and then four hydraulic cylinders, pushing from below, locate the upper surface of the pallet against hardened and ground horizontal stops.

Horizontal guide bars of the spindle head enter the guide bushings *5* in the pallet uprights.

The lower surface of the pallet is subject to wear as it slides along the guide strips, while the upper surface retains its accurate location. This substantially lengthens cutting tool life.

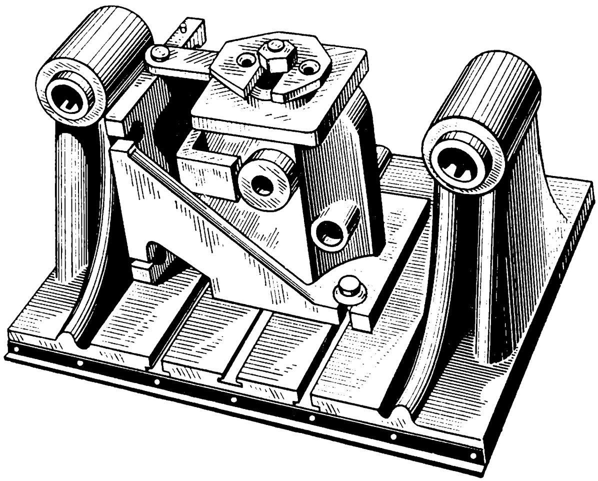


Fig. 249. A workpiece clamped in a holder mounted on the pallet

Holder *6* for one of the different parts has a bushing *8* and a key *7* enabling it to be readily aligned with central pin *4* and keyway *3*.

To machine the same housing-type part in different positions—vertically, as shown in Fig. 249, or horizontally—the holder has two datum surfaces and is reinstalled on the pallet to pass again through all the stations after the transfer machine has been changed over.

Two setters-up change over this transfer machine, including the replacement of 12 multiple-spindle heads and 12 workpiece holders, in 5 hours.

CHAPTER 18

OUTPUT AND STRUCTURE OF TRANSFER MACHINES

18-1. Output of Transfer Machines

The design piece output Q_t of a single-flow transfer machine is

$$Q_t = \frac{1}{\tau} \text{ pcs per min} = \frac{60}{\tau} \text{ pcs per hour} \tag{82}$$

where τ = piece time and is the design pace of the transfer machine, i.e., the transfer machine cycle time or the time intervals after which successive finished workpieces are unloaded at the end of the transfer machine or line

$$\tau = T_{c\,max} + t_{hl} \text{ min} \tag{83}$$

where $T_{c\,max}$ = machine cycle time of the limiting component machine tool in the transfer machine or line, min

t_{hl} = duration of the unoverlapped handling operations in the transfer machine or line (workpiece transfer time, time required for the operation of an automatic operator, for location, clamping, unclamping, delocation, etc.), min.

The actual piece output or production rate Q_a is less than the design value due to noncyclic time losses. Thus

$$Q_a = Q_t - q_l = Q_t \eta_l = \frac{60}{t} \text{ pcs per hour} = \frac{1}{t} \text{ pcs per min} = \frac{1}{\tau}\,\eta_l \tag{84}$$

where q_l = loss in output of the whole transfer machine or line per unit of time, pcs

t = length of the average interval after which finished workpieces are unloaded from the last station of the transfer machine or line, min

η_l = utilization factor of the transfer machine or line.

It is evident from equation (84) that

$$\eta_l = \frac{Q_t - q_l}{Q_t} = \frac{Q_a}{Q_t} = \frac{\tau}{t} \tag{85}$$

The relative fraction β_l of piece output loss of the transfer machine is

$$\beta_l = \frac{Q_t - Q_a}{Q_t} = 1 - \eta_l \tag{86}$$

The loss in piece output q_l for the transfer machine or line is

$$q_l = Q_t - Q_a = (1 - \eta_l)\, Q_t = Q_t \beta_l \tag{87}$$

The piece output loss q_l and the technical utilization factor η_l of the transfer machine depend upon: (a) the frequency with which troubles occur in the elements of the transfer machine equipment leading to downtime of certain units due to their own malfunctioning (self-inflicted downtime); (b) the time required to remedy the troubles of the units due to their own malfunctioning, and (c) downtime of the units, due not to their own malfunctioning, but to rigid interlocking in the working cycle with another unit in which the trouble occurs (this is sometimes called "imposed downtime").

The last type of downtime is manifested to the fullest extent in single-flow interlocked transfer machines.

18-2. Losses and Technical Utilization Factor of Single-Flow Interlocked Transfer Machines

Upon trouble in one of the component units in an interlocked transfer machine, all the other units of the machine stop operation and during this time no trouble can occur in them. Therefore, self-inflicted downtime periods of the component units of an interlocked transfer machine (downtime to eliminate malfunctioning of one of the elements of the unit) occur only consecutively and cannot overlap in time.

Let us denote the self-inflicted downtime, referred as an average to a unit of actual operating time, by D_{s1} for the first component unit, by D_{s2} for the second, D_{s3} for the third, etc.

The average full time loss (from self-inflicted and supplementary downtime) D_{f1} of each unit of the transfer machine per unit of operating time is

$$D_{f1} = \sum_{i=1}^{N} D_{si} \tag{88}$$

where N is the total number of interlocked component units in the transfer machine.

The design piece output Q_t is the same for all working units of the interlocked transfer machine but, due to losses, Q_t workpieces are machined in each unit in $1 + \sum_{i=1}^{N} D_{si}$ units of time. Thus, the actual piece output of the transfer machine is

$$Q_a = \frac{Q_t}{1 + \sum_{i=1}^{N} D_{si}} \tag{89}$$

The technical utilization factor η_{int}, not taking into account the losses due to organizational delays, is expressed by the relationship

$$\eta_{int}=\frac{Q_a}{Q_t}=\frac{1}{1+\sum_{i=1}^{N} D_{si}} \tag{90}$$

From this it follows that for the simplest structural scheme of interlocked transfer machines (single-flow and without stockpile banks), the *general (overall) losses of the transfer machine are equal to the sum of the self-inflicted losses of all the component units.*

The individual utilization factor of a component unit is

$$\eta_{si}=\frac{1}{1+D_{si}} \tag{91}$$

and the self-inflicted downtime of this unit (due to its own malfunctioning) per unit of its operating time is

$$D_{si}=\frac{1-\eta_{si}}{\eta_{si}} \tag{92}$$

Consequently, the technical utilization factor of an interlocked transfer machine can also be expressed as

$$\eta_{int}=\frac{1}{1+\sum_{i=1}^{N}\frac{1-\eta_{si}}{\eta_{si}}} \tag{93}$$

Equations (88) and (90) indicate that the overall losses of an interlocked transfer machine (per unit of its operating time) are equal to the average length of time of the full losses (from self-inflicted and supplementary downtime) of each component unit referred to a unit of its operating time.

The supplementary losses of each component unit of an interlocked transfer machine increase with N, the number of component units. For this reason, interlocked transfer machines or their sections are not designed with a large number of component machine tools, rarely more than ten for transfer machines handling housing-type parts. Efforts are also made to reduce the amount of auxiliary units. On the other hand, each component unit of integrated construction is interlocked and the formulated thesis concerning the self-inflicted and supplementary downtime of its subassemblies and constructional elements is valid within the limits of the component unit, as well as for the whole transfer machine.

Hence, to increase the utilization factor of interlocked transfer machines, it is of prime importance to reduce, as much as possible, the number of subassemblies and constructional elements, both within the limits of the compo-

TABLE 6

Self-Inflicted Losses of Equipment in Transfer Machines

Name of device	Number of stoppages per 100 min of operation	Average length of downtime, min, per 100 min of operation
Power unit	0.009	0.42
Clamping device:		
of medium complexity	0.003	0.10
with complex clamping facilities	0.005	0.20
Pallet with simple workpiece clamping	0.001	0.01
Device for automatically clamping workpieces in the pallets	0.002	0.09
Turning device	0.002	0.06
Pusher or lifting device for handling pallets	0.001	0.02
Pallet return conveyer	0.002	0.06
Turnover device for dumping chips out of machined holes	0.002	0.05
Device for checking the depth of drilled holes	0.003	0.09
Devices common to the whole section of the transfer machine (transfer bar, hydraulic drive of the transfer bar and clamping device, electrical equipment)	0.05	0.75

nent units themselves (common drive for many spindles and their common feed facilities in unit-built machines), and over the extent of the whole transfer machine. Examples are the common feed drive of the milling heads and the transfer bar powered by a single hydraulic cylinder (Fig. 222) and the drive of a transfer bar from the table of a planer-type milling machine (Fig. 230). Such measures prove to be especially efficient when dependable synchronization of the operation of the component units is achieved by simple and reliable mechanical interlocking devices, and subassemblies with less dependable constructional elements are eliminated.

Under the conditions of the cutting speeds and feeds applied in transfer machines, the self-inflicted losses of the machine tools, handling devices and other equipment do not depend to any practical extent upon the load. This enables the self-inflicted losses of the equipment to be determined on the basis of observational data on transfer machine operation. Examples of such observations are listed in Table 6.

As an example, making use of Table 6, we can determine the losses due to the malfunctioning of equipment for an interlocked transfer machine made up of 16 power units, eight clamping devices of medium complexity, 18 pallets (fixtures), two devices for clamping and releasing the workpieces in the

pallets, two lifting devices for handling pallets, one pallet return conveyer, one fixture for checking the depth of drilled holes, and one set of devices common to the whole transfer machine (or its section).

The overall general losses D_{int} of an interlocked transfer machine or line are equal to the full amount of time lost by each of its component units. Thus

$$D_{int} = D_{f1} = \sum_{i=1}^{N} D_{si} = 0.42 \times 16 + 0.1 \times 8 + 0.01 \times 18 + 0.09 \times$$
$$\times 2 + 0.02 \times 2 + 0.02 \times 2 + 0.06 \times 1 + 0.09 \times 1 + 0.75 \times 1 =$$
$$= 8.82 \text{ min per 100 min of transfer machine operation}$$

Time losses due to cutting tools depend upon the cutting speed and feeds and the depth of cut. These, in their turn, determine the tool life (period of operation between resharpenings) and $R_{i\,max}$, the maximum number of workpieces machined between grinds (sharpenings) for each group of cutting tools (see Sec. 16-5). The time required per 100 min of operation to change a group of tools is

$$D_{ti} = \frac{(t_{i\,del} + t_{i\,ch}\, m_i)\frac{Q_t}{60}100}{R_{i\,max}} \text{ min} \tag{94}$$

where Q_t = design piece output, pcs per hour

$t_{i\,del}$ = time required to deliver a group of cutting tools to the machine tool, min

$t_{i\,ch}$ = time required to change one tool of the given group, min

m_i = number of tools in the given group.

Thus, for a group of $m_i = 10$ twist drills from 8 to 25 mm in diameter, for which $R_{i\,max} = 380$ pcs, $t_{i\,del} = 1$ min, $t_{i\,ch} = 0.2$ min and $Q_t =$ $= 40$ pcs per hour (determined by time-study techniques),

$$D_{ti} = \frac{(1 + 0.2 \times 10)\frac{40}{60}100}{380} \cong 0.53 \text{ min}$$

per 100 min of transfer machine operation.

Along with downtime due to regular dulling of the cutting tools, accidental downtime may occur due to malfunctioning, premature wear and breakage of the tools. Approximate data on accidental time losses due to cutting tool troubles are listed in Table 7.

In the case of a group of $m = 10$ twist drills, for which $R_{max} = 380$ pcs between resharpenings and $Q_t = 40$ pcs per hour, the accidental time losses are

$$D_{t\,ac\,i} = \frac{0.12\, m \frac{Q_t}{60} 100}{R_{max}} = 0.12 \times 10 \times \frac{40}{60} \times \frac{100}{380} = 0.21 \text{ min} \tag{95}$$

per 100 hours of transfer machine operation.

TABLE 7

Accidental Downtime Due to Cutting Tool Troubles

Type of cutting tool	Average number of accidental stoppages during the tool life	Average length of accidental downtime, min, during the tool life
Twist drill of small diameter or large length-to-diameter ratio	0.03	0.18
Twist drill of medium size	0.01	0.12
Tap of small diameter	0.05	0.30
Tap 8 to 25 mm in diameter	0.03	0.27
Core drill, counterbore, spotfacer	0.02	0.18
Reamer	0.015	0.18
Cutter clamped in a boring bar:		
roughing cutter	0.02	0.20
finishing cutter	0.015	0.18

After calculating the regular and accidental time losses for all the groups of tools, relating them to 1 min of operation time and adding them to the time losses due to malfunctioning of the equipment of the transfer machine or line (also related to 1 min of operation time), the technical utilization factor of an interlocked transfer machine can be determined from the formula

$$\eta_{int} = \frac{1}{1 + \sum_{i=1}^{N} D_{si} + \sum_{i=1}^{M} D_{ti} + \sum_{i=1}^{M} D_{t\,ac\,i}} \tag{96}$$

where M is the number of groups of cutting tools.

The time-study data on downtime due to equipment and cutting tool troubles listed in the tables are conditional to some degree but they enable various design versions of a transfer machine to be compared in respect to their technical utilization factor.

18-3. Dividing an Interlocked Transfer Machine into Flow Lines

An exceptionally long limiting operation, as well as a high rate of production, may require that several workpieces be machined in parallel at the limiting operations. This can be done in two ways.

1. The machine tools for the limiting operation are installed in a single file.

If the workpieces are small, several can be machined in parallel in a single machine tool (Fig. 251). Large workpieces can be machined in parallel, one

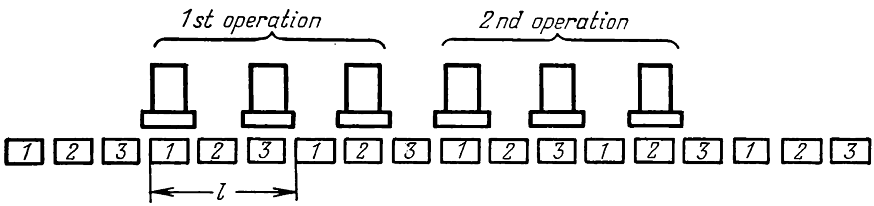


Fig. 250. Layout for machining several large workpieces in parallel in single file

in each machine tool (Fig. 250). In both cases, if z workpieces are machined in parallel, the pitch, or stroke, of the transfer mechanism is taken to be z times the distance between adjacent workpieces. Idle stations are provided between the machine tools.

When machining is carried out in parallel in a single flow line, the number of interlocked machine tools is increased. This reduces the technical utilization factor of the interlocked transfer machine. Consequently, this method of parallel operation is not suitable for a transfer machine with a large number of component machine tools.

2. Machine tools that machine several workpieces in parallel are installed in parallel parts of the transfer machine equipped with handling devices that enable the parts to operate independently of one another. As a result, the technical utilization factor is substantially raised in comparison with that of an ordinary interlocked transfer machine. These parallel parts are called flow lines.

In each of the four flow lines of the transfer machine for cylinder blocks of tractor engines shown in Fig. 252, the main bearings of the crankshaft and the camshaft are finish bored, the bearing lock is faced and the dimensions are checked.

Each flow line is equipped with three transfer bars: A—for approach, B—for feeding the cylinder blocks into the flow line, and C—for delivering

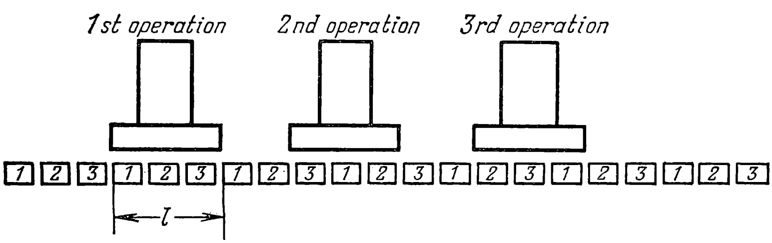


Fig. 251. Layout for machining several small workpieces in parallel in single file

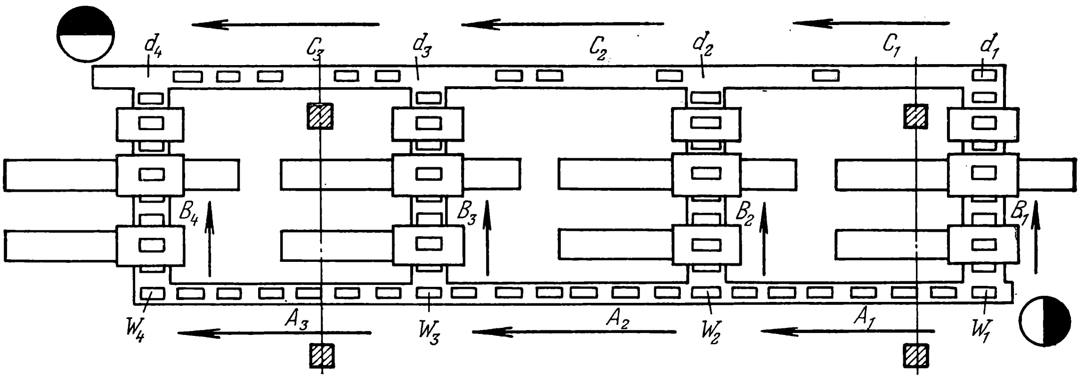


Fig. 252. Transfer machine incorporating four flow lines for machining the cylinder blocks of tractor engines

the blocks. The flow lines are interlocked tunnel-type transfer machine sections. The handling system of the transfer machine operates as follows.

If there is a workpiece at station W_1, machining is completed in flow line *1* and delivery station d_1 is free, transfer device B_1 will operate. If any one of these conditions is not complied with and station W_2 is occupied, the workpiece remains stationary at station W_1.

If, however, station W_2 is free, transfer device A_1 will operate and the workpiece will be delivered to station W_2. Under conditions analogous to those given above for flow line *1*, the workpiece is either fed into flow line *2*, remains stationary or is carried by transfer device A_2 to station W_3.

The third and fourth flow lines are controlled in an identical manner. Delivery transfer devices C_1, C_2 and C_3 are operated in accordance with a corresponding set of conditions.

In an interlocked transfer machine made up of p parallel flow lines operating independently of one another, self-inflicted downtime of the elements of equipment occurs consecutively only within the limits of each separate flow line, i.e., within the limits of $\frac{1}{p}$ of all the units of equipment involved. Consequently, the full time loss (the sum of the self-inflicted and supplementary losses, the latter being the sum of the self-inflicted losses of the elements of equipment of each separate flow line) of each flow line is only $\frac{1}{p}$ of that for a single-flow interlocked transfer machine. Since the overall general losses D_{int} of an interlocked transfer machine are equal to the full losses (sum of the self-inflicted and supplementary losses) of each component element, the losses in a system with p parallel flow lines are only $\frac{1}{p}$ of those

of a single-flow interlocked transfer machine having exactly the same equipment. The technical utilization factor is increased when a transfer machine is divided into flow lines. Thus

$$\eta_l = \frac{1}{1 + \sum_{i=1}^{\frac{N}{P}} D_{si}} = \frac{1}{1 + \frac{D_{int}}{p}} \tag{97}$$

where the notation is the same as in equation (88).

Losses in the approach and delivery transfer devices (Fig. 252) can be taken into consideration by introducing a loss increase factor with a value of the order of 1.01 or 1.02.

In multiple-flow line transfer machines stoppages may occur simultaneously in several flow lines. Hence, to avoid downtime due to delays caused by waiting for setters-up and other maintenance men busy at other flow lines, the number of persons servicing such transfer machines should be greater than for single-flow arrangements.

Cross conveyers are required in transfer machines for housing-type parts (see Fig. 218) at the point of transition from a multiple-flow section to a single-flow section, or from one multiple-flow section to another with a different number of flow lines.

18-4. Dividing Transfer Machines for Housing-Type Parts into Sections

If a complex workpiece is to be machined in a great number of consecutive stations of an interlocked transfer machine, losses can be reduced and the technical utilization factor can be raised by dividing the machine into consecutive sections between which stockpiles of qualified parts are provided. Such stockpiles are also provided if there is a multiple-flow section in an interlocked transfer machine.

The simplest stockpile banks for housing-type parts have the form of storage areas, located at the junctions of the transfer machine sections and manually serviced.

Two types of automated stockpile banks for housing-type parts are applied: *progress-through* and *blind-alley* banks.

The distinguishing feature of progress-through banks is that the workpieces are transferred into them, not only during a stoppage of an adjacent section, but also during normal operation. Workpieces are not held for long periods of time in such banks and are not subject to corrosion, but, on the other hand, troubles in the stockpile bank lead to downtime of the adjacent sections of the transfer machine. For this reason, the conveyer between successive sections is usually used as a progress-through stockpile bank for housing-type

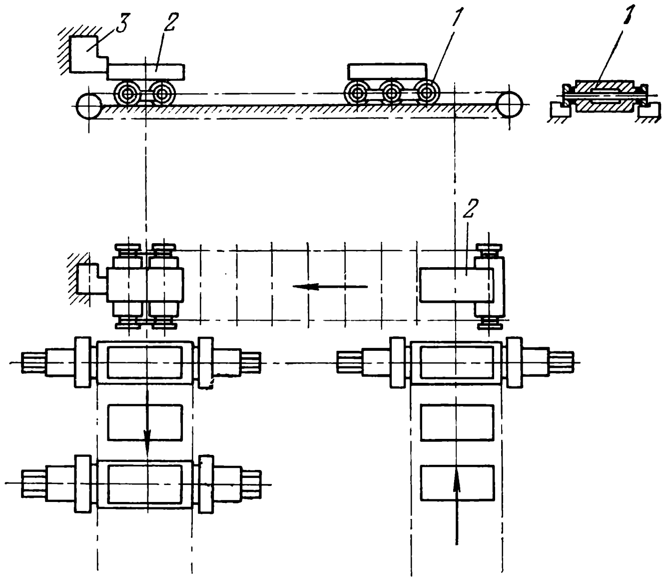


Fig. 253. Principle of a progress-through stockpile bank for housing-type parts

parts (Fig. 253). The conveyer in this case is designed as an endless chain of freely rotating rollers *1*. When part *2* runs up against stationary stop *3* or against a previously delivered part, the rollers roll along the undersurface of the stationary part.

Blind-alley stockpile banks operate only during the downtime of one of the adjacent sections. Therefore, malfunctioning of the bank does not lead to supplementary downtime of the sections when they operate together normally (Fig. 254).

In this arrangement, section *8* delivers the workpieces to station *6* which is linked to the cross conveyer *7* as well as to conveyer *5* which is intended for delivering workpieces into and out of the blind-alley stockpile bank. The bank is designed as racks with skids (guides) along which the workpiece is conveyed by transfer bars with two rows of disappearing fingers facing opposite directions (Fig. 255). Conversion from workpiece delivery into the bank to delivery out of it takes place when the transfer bar is turned 180° about its axis. The presence of workpieces at stations *2* (Fig. 254) and *6* is checked by means of limit switches. The conveyer of section *8* will deliver a workpiece to station *6* only if it (the station) is free.

The conveyer of section *7* will advance only if there is a workpiece at station *6* (which it picks up) and none at station *2*.

The conveyer of section *1* will advance (at the end of the working cycle in section *1*), picking up a workpiece at station *2*, only if such a workpiece is available at this station.

These conditions are complied with if the sections operate normally and in strict co-ordination. In the case of a stoppage in section *8*, stations *6* and *2* are free (conveyer *7* and section *1* are stopped down). Now, if there is at least one workpiece available on the first conveyer *3* of the stockpile (this being established by means of a limit switch), conveyer *3* feeds out the workpiece and conveyer *5* delivers it to station *6*. This switches on conveyer *7* and section *1*. When section *8* resumes operation, workpieces are no longer fed out of the stockpile. On the other hand, if section *8* does not resume operation, all the workpieces in the stockpile are used up, after which all the conveyers of sections *8* and *1* and those in the stockpile are switched off.

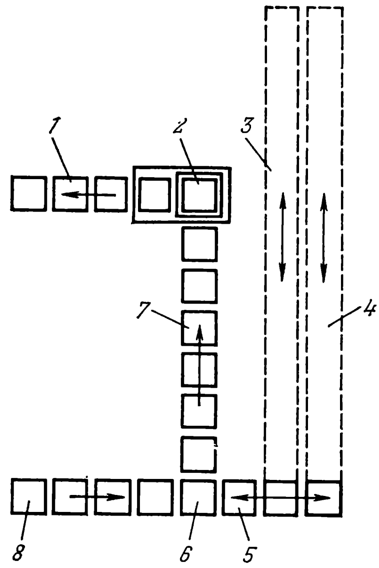


Fig. 254. Principle of a blind-alley stockpile bank for housing-type parts

If, during normal operation, the workpiece at station *2* is not picked up by section *1* and there is still some free space in the stockpile, conveyer *5* picks up the workpiece at station *6* and the transfer bar of the stockpile begins to operate with its feed-in function, continuing until section *1* resumes operation or until all the free space is occupied in the stockpile.

A transfer machine with a blind-alley stockpile bank is shown schematically in Fig. 256. It was designed for machining engine blocks in 12 unit-built machine tools: three each in sections *I* and *II*, and six in section *III*. Two cylinder blocks, held in a two-place fixture, are machined simultaneously in each machine tool. The transfer machine is divided into sections at the points where turning devices are installed.

In an in-line layout of a transfer machine for housing-type parts, the stockpile banks, designed in the form of reserve conveyers, can be arranged

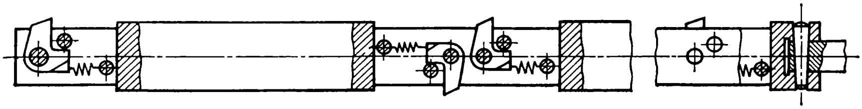


Fig. 255. Transfer bar of a stockpile bank conveyer

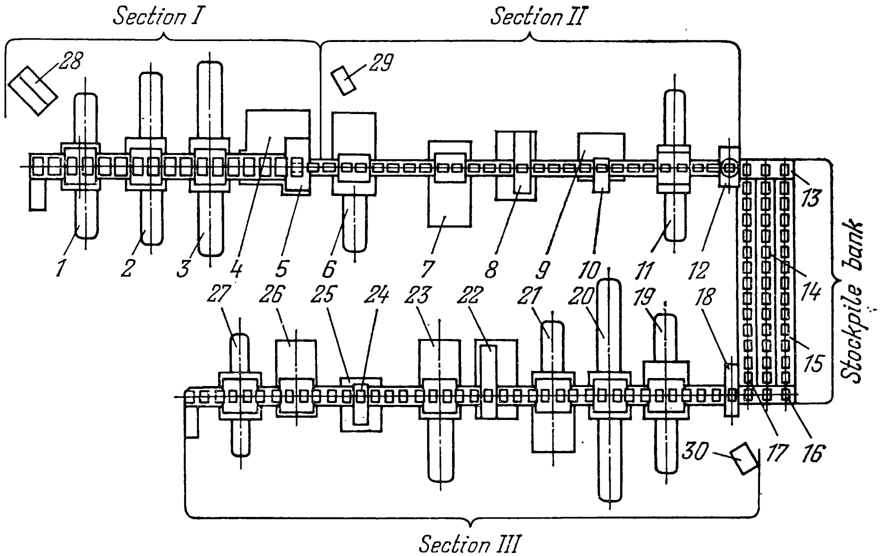


Fig. 256. Transfer machine for engine blocks:

1, *2* and *3*—component machine tools of section *I*; *4*, *9* and *25*—hydraulic power stations; *5* and *18*—turning drums; *6*, *7* and *11*—component machine tools of section *II*; *8* and *22*—chip dumpers; *10* and *24*—inspection fixtures; *12*—turntable; *13*, *14*, *15*, *16* and *17*—stockpile bank conveyers; *19*, *20*, *21*, *23*, *26* and *27*—component machine tools of section *III*; *28*—general control desk of the transfer machine; *29* and *30*—control desks of sections *II* and *III*

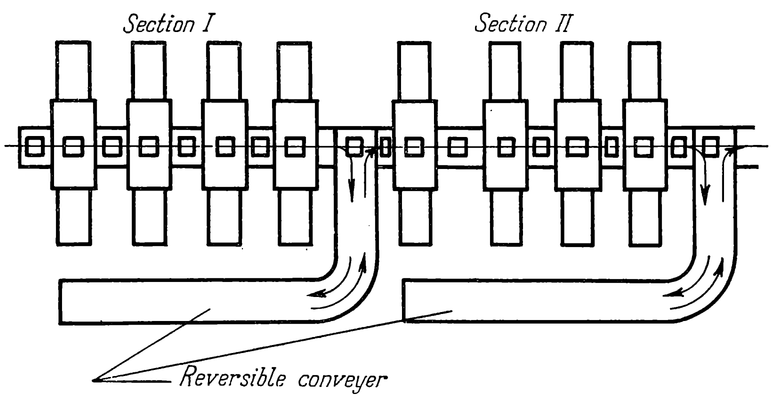


Fig. 257. Layout in which stockpile banks for housing-type parts are arranged parallel to the main transfer line

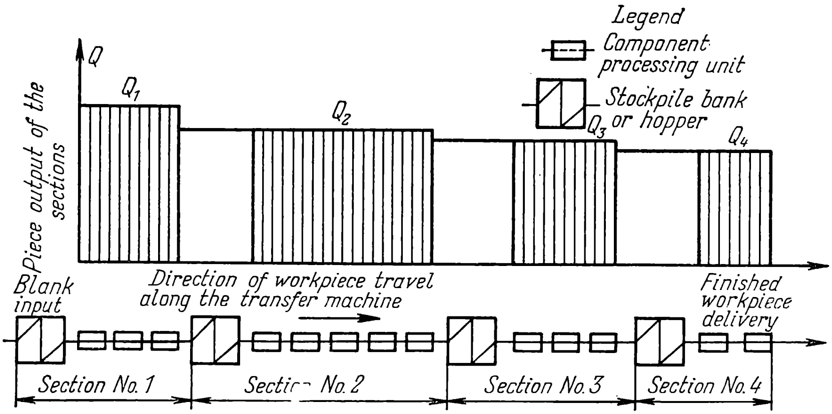


Fig. 258. Diagram showing the variation in output of the sections of a transfer machine

parallel to the main transfer line of the machine (Fig. 257). In the process of transfer machine operation, the number of parts in the stockpile may vary from zero to the maximum capacity. To keep the stockpiles from being depleted, the average output of the sections between the stockpile banks should increase toward the point where machining begins (Fig. 258).

The division of the transfer machine into sections and the selection of the locations of the stockpile banks are most expedient when the amounts of the downtime due to malfunctioning, and the times required to remedy the troubles, are equal in adjacent sections. In any other principle of division, the overall losses of the transfer machine will be equal to the length of downtime of the section with the highest losses, and the sections with lower losses will be subject to additional downtime. In this case, a difference of 30 to 40 per cent in the length of downtime of two adjacent sections will not significantly affect the value of the technical utilization factor η_l of the transfer machine or line. For this reason, stockpile banks are most often located at transition points in the sequence of processing operations.

In dividing an interlocked transfer machine having a total average downtime D_{int} per unit of operating time into m sections with $(m-1)$ stockpiles having average lengths of self-inflicted downtime D_{st} per unit of operating time, the same for each stockpile, the average length of downtime D_l of the divided transfer machine per unit of time is

$$D_l = D_{int}\frac{1+(m-1)\sigma}{m} + D_{st}(m-1) \qquad (98)$$

where σ is the factor of imposed losses which takes into account the increase in the losses of each section due to downtime of an adjacent section caused by depletion of the stockpile or its overfilling.

It is assumed, in this case, that the interlocked transfer machine is divided into sections having an equal average number of troubles U_i occurring in a unit of time, and with an equal average specific length of downtime B_i of each section required to remedy the trouble per unit of time. Such a division is optimum in respect to the overall losses of the divided transfer machine. Research in this field has shown that the factor of imposed losses σ, in this case, is determined by the formula

$$\sigma = \frac{1}{1 + \frac{ktz_m}{2}} \tag{99}$$

where t = average length of operation of each section of the transfer machine, min

z_m = capacity of a stockpile bank, pcs

$k = k_i = \frac{U_i}{B_i}$ = reciprocal of the average length of time in minutes required to remedy one trouble.

Using equation (98), we can determine the coefficient μ of the reduction in losses when an interlocked transfer machine is divided into sections. Thus

$$\mu = \frac{D_l}{D_{int}} = \frac{1 + (m-1)\,\sigma}{m} + \frac{D_{st}\,(m-1)}{D_{int}} \tag{100}$$

Finding the extreme of this function $\left(\frac{d\mu}{dm} = 0\right)$, we can determine the optimum number of sections for a single-flow transfer machine in respect to the minimum overall losses of a divided arrangement. Thus

$$m = \sqrt{\frac{D_{int}}{D_{st}}\,(1-\sigma)} \tag{101}$$

If $\frac{D_{int}}{D_{st}} = 90$ and $\sigma = 0.3$, the optimum number of sections $m = 8$.

The general theory of transfer machines has not yet been fully worked out, nor has experience gained in the operation of such machines been sufficiently generalized. Certain investigations have led to the conclusion that, for a given sum of the capacities of the stockpile banks, the more the number of sections, the less the losses in the system. Therefore, to reduce losses, it would be desirable to provide stockpile banks between each two adjacent machine tools, as is done in an unautomated production line.

A reduction of the losses is not the only factor to be considered in dividing a transfer machine into sections. Of essential importance is the cost of the stockpile banks. A stockpile bank can be efficiently introduced between adjacent machine tools if this does not involve large expenditures, and if it is combined with the part transfer or orientation system (for small parts). Sometimes stockpiles are not used at all because of the large expenditures

they lead to. A palletized transfer machine is not commonly divided into sections, as various complications ensue and the number of pallets must be substantially increased.

The required capacity of a stockpile bank increases with the average length of downtime of the section and, consequently, with the average length of the self-inflicted downtime of the elements of the equipment.

Theoretically, it is recommended that the stockpile bank or hopper capacity be taken as at least tenfold the amount of workpieces that are accumulated (or expended) during a stoppage of average length. In transfer machines for large housing-type parts, with average values of a single self-inflicted downtime ranging from 20 to 30 min, the capacity of the stockpile banks is usually taken as the number of workpieces required during 30 to 120 min of operation.

CHAPTER 19

TRANSFER MACHINES FOR SHAFT PRODUCTION

The end faces of shaft blanks are milled and the centre holes are drilled not only in single-station double-end machines but, if the volume of production is sufficiently large, in six-station drum-type milling and centring machines as well (for example, Soviet model MP-78).

In recent years, it has been found preferable to turn stepped shafts in semiautomatic hydraulic tracer-controlled lathes, models MP-106, 1712, 1722, etc., instead of using multiple-tool semiautomatics for this purpose. Hydraulic tracer-controlled semiautomatics possess numerous advantages over multiple-tool models: longitudinal turning is carried out with a single tool, thereby reducing setting-up time and tool costs and allowing the lathe to be equipped with an automatic tool-setting device operating on the feedback principle; shafts can be turned to a diametral tolerance of 0.1 or 0.15 mm, instead of the 0.3 or 0.4 mm attainable in multiple-tool semiautomatics.

The continuous chip, sometimes of considerable length, obtained in single-tool turning can lead to difficulties. This factor raises the requirements made to the layout of the lathe and its slides in regard to chip disposal and to the accessibility of the cutting zone and the chip conveyer for observation and for trouble-shooting in connection with chip disposal.

Shafts are usually ground in semiautomatic cylindrical grinders and centreless grinders. The convenience of locating a shaft between centres and clamping it in a self-clamping driver chuck considerably simplifies the construction of the required clamping devices Another favourable factor in handling shafts is the possibility of precise orientation in V-blocks.

Depending upon the layout of the machine tools (carriage and slide units of lathe-type machines), one of three main systems is used for handling the shafts being produced in a transfer machine: overhead, frontal and through-horizontal transfer systems.

An *overhead system* with top loading of the workpiece (Fig. 259) is used when the component lathe-type machine tools are designed with a horizontal layout (arrangement of the units). This concerns semiautomatic lathes for machining crankshafts, camshafts and other like parts that are turned with cross feed of the tools and when no long continuous chips are produced.

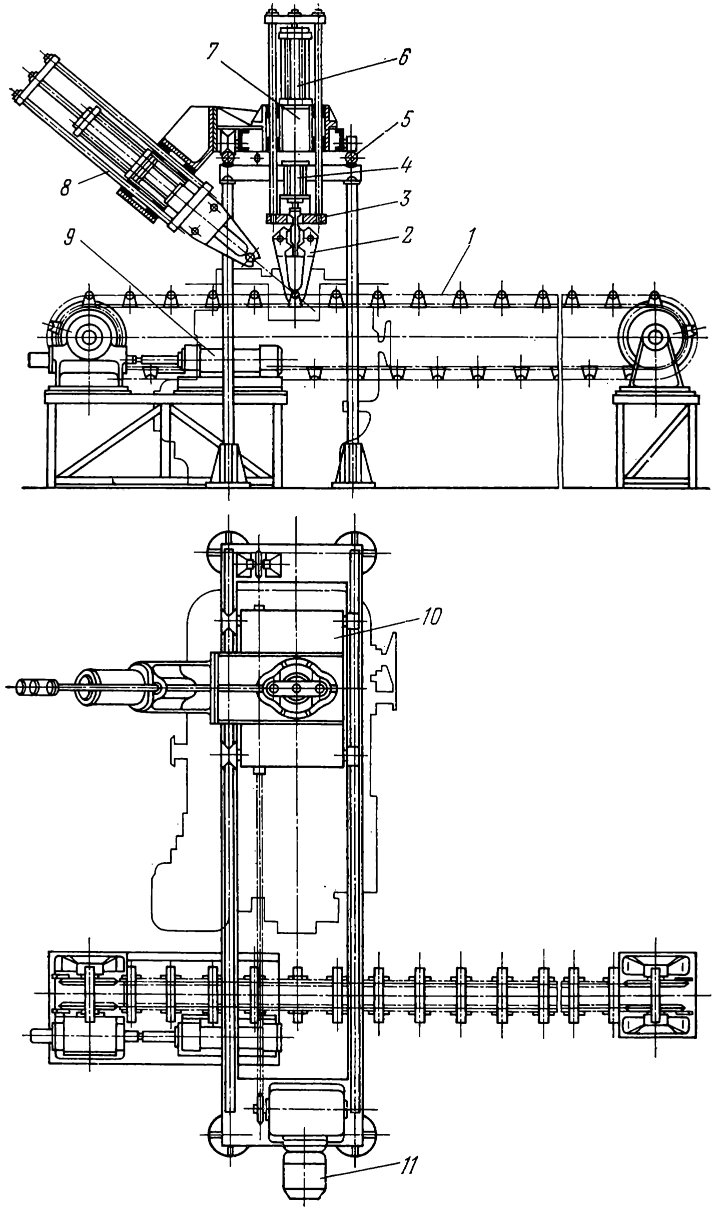


Fig. 259. Top loading of shafts into a multiple-tool semiautomatic lathe:

1—chain conveyer with V-blocks; *4*—hydraulic cylinder for closing the tongs *2* to grip the workpiece; *5*—guide rails; *6*—cylinder for raising and lowering cross-member *3* with tongs *2*; *7*—lifting device for blank loading; *8*—lifting device for unloading finished workpieces; *9*—hydraulic cylinder for advancing conveyer *1* by one pitch; *11*—electric motor of the reducing gear for traversing carriage *10* by means of an endless chain

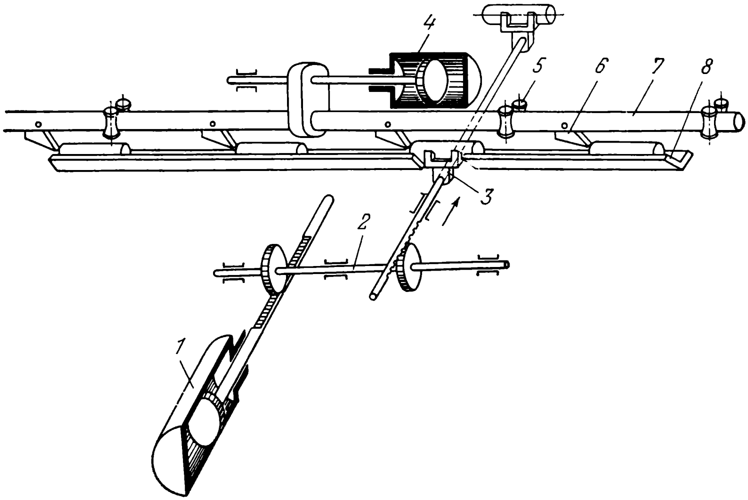


Fig. 260. Principle of the handling system of the transfer machine for producing electric motor shafts:

1—hydraulic cylinder for actuating the feeders; 2—common shaft of the feeder drive; 3—feeder grip; 4—hydraulic cylinder for actuating transfer bar 7; 5—roller support of transfer bar 7; 6—disappearing fingers of transfer bar 7, which push the shafts along trough 8 and the V-blocks of feeder grips 3

Top loading of the workpiece proves convenient for general-purpose engine lathes and their modifications. Such lathes, however, cannot be readily built into a transfer machine for shaft production, especially if tracer-controlled slides are to be used, since the layout of these slides is not easily adapted for automatic chip disposal from the cutting zone.

A *frontal system* with side horizontal loading is used for vertical semiautomatic lathes, as well as for horizontal semiautomatics having their carriages arranged either vertically or at an angle to the vertical.

A *through-horizontal system* of shaft transfer through the clamping position is suitable only for vertical or inclined carriage arrangement in semiautomatics with a frame-type layout (for example, models 1712 and 1722).

All of the first transfer machines built in the USSR for shaft production had the same distinguishing feature: they were equipped with a common transfer bar which pushed the shafts along a common trough with gaps for crosswise operation of trough, or V-block, feeders (Fig. 260).

Single-grip feeders (Fig. 261) of these handling systems all operated simultaneously according to the following cycle: the machine stops and the feeder advances to remove the finished workpiece, the workpiece is unclamped and gripped by the feeder, the feeder carries the workpiece back to the trough

line, the feeder grips a new workpiece and carries it to the line of centres where it is clamped, the machine starts and the feeder retracts.

During the time that the workpiece is being carried from the machine tool to the conveyer, that it is advanced by the transfer bar and a new workpiece is carried to the clamping position, the machine stands idle. This is the chief disadvantage of this system of handling.

These transfer machines had no stockpile banks. This led to excessive time losses during downtime due to malfunctioning or in changing over the transfer machine for a different size of shaft.

Later two-grip feeders were incorporated in the design of these transfer machines. This changed the cycle of operation of the handling system (Fig. 262) and the feeders (Fig. 263), reducing the time loss in removing the finished workpiece and in loading a new one.

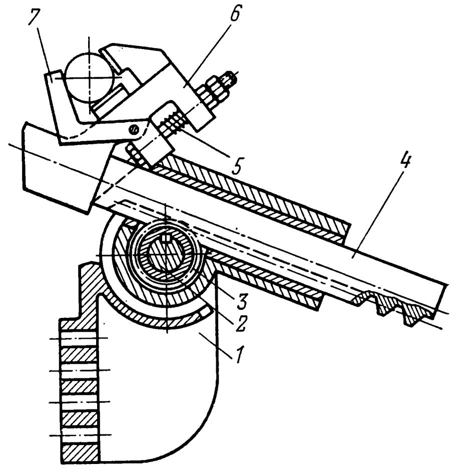


Fig. 261. Single-grip feeder:
1—body of the feeder; *2*—feeder drive shaft; *3*—pinion; *4*—rack; *5*—springs holding the shaft blank; *6*—V-block trough at the end of the rack; *7*—lever actuated by springs *5*

In connection with the use of two feeder grips which remove the workpiece from the trough and return it to the trough at different heights, each machine tool has a separate trough inclined downward in the direction of workpiece transfer (Fig. 264).

Time losses due to downtime during trouble-shooting and during changeovers for producing shafts of a different size have been reduced in the model Э-200 transfer machine by providing a model Э-204 stockpile bank following the group of model MP-106 lathes (Fig. 264).

In the model Э-200 transfer machine for the production of spline shafts, four cylindrical grinders, models ЭВ153 and ЭШ153. have been equipped with feedback-circuit in-process gauging systems. A system of selective inspection is used for all the other machine tools in which the operator checks workpieces in measuring instruments installed alongside the corresponding machine tools.

In a transfer machine for producing electric motor shafts, each model 1712 semiautomatic lathe is followed by an automatic gauging device which carries out 100-per-cent inspection of the shafts and transmits commands for readjusting the tools whenever necessary.

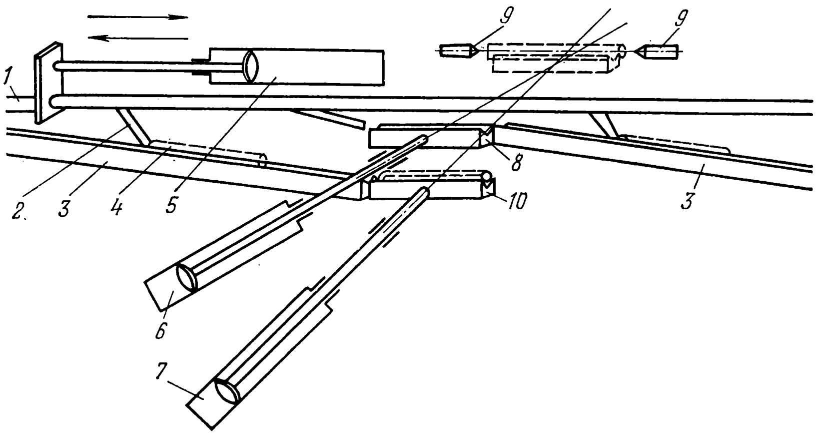


Fig. 262. Principle of a handling system with two-grip feeders:
1—transfer bar; *2*—transfer bar finger; *3*—inclined trough of the transfer system; *4*—workpieces in the troughs; *5*—hydraulic cylinder for actuating the transfer bar; *6*—hydraulic cylinder of the grip for removing the finished workpieces; *7*—hydraulic cylinder of the grip for loading the blanks; *8* and *10*—V-blocks of the grips; *9*—lathe centres

In the model Э-200 transfer machine for spline shaft production (Fig. 264) the chips are removed from the model MP-106 hydraulic tracer-controlled semiautomatics and the model MA-4B spline planing machines by screw-type chip conveyers. Chips are removed manually by the operators at all the other machine tools at the end of the shift.

The sequence of operations in these transfer machines is controlled by the in-travel principle. If any operation in the sequence has not been carried out, a trouble signal lights up on the control desk and the transfer machine stops automatically.

19-1. Transfer Machine, Model MP-107

Through-horizontal shaft transfer through the clamping zone has been applied in the design of the model MP-107 transfer machine which consists of two hydraulic tracer-controlled lathes (modifications of the model 1712 semiautomatic). The lathes are installed one opposite the other to enable shafts up to 90 mm in diameter and up to 380 mm long to be machined from both ends without being turned end for end.

The shaft blanks are loaded into a chain-type magazine with a capacity sufficient for 30 min of transfer machine operation. Magazine chain *1* is driven by electric motor *10* (Fig. 265) until the next blank reaches the initial position in which it is gripped by the loading conveyer and where it

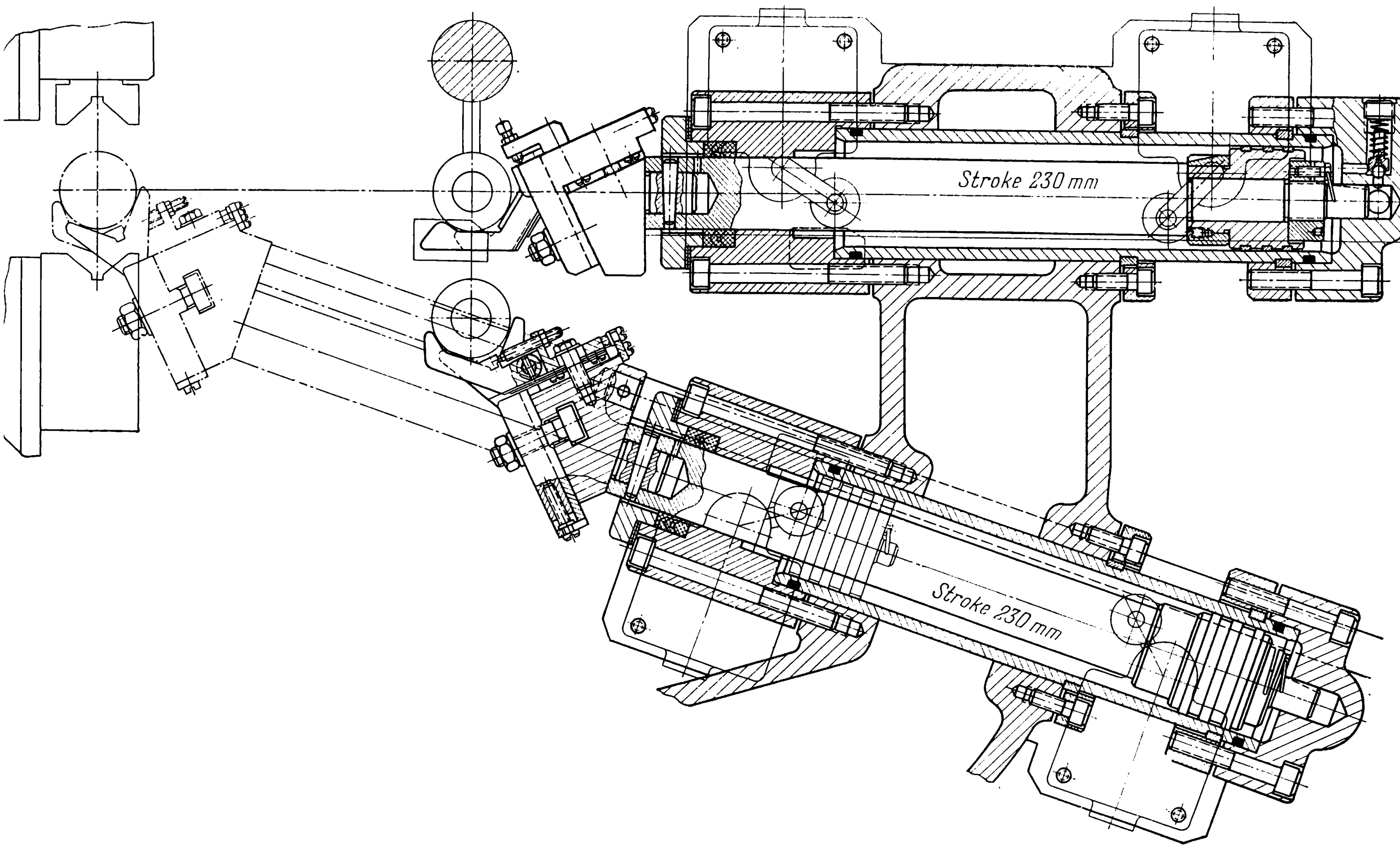


Fig. 263. Two-grip feeder

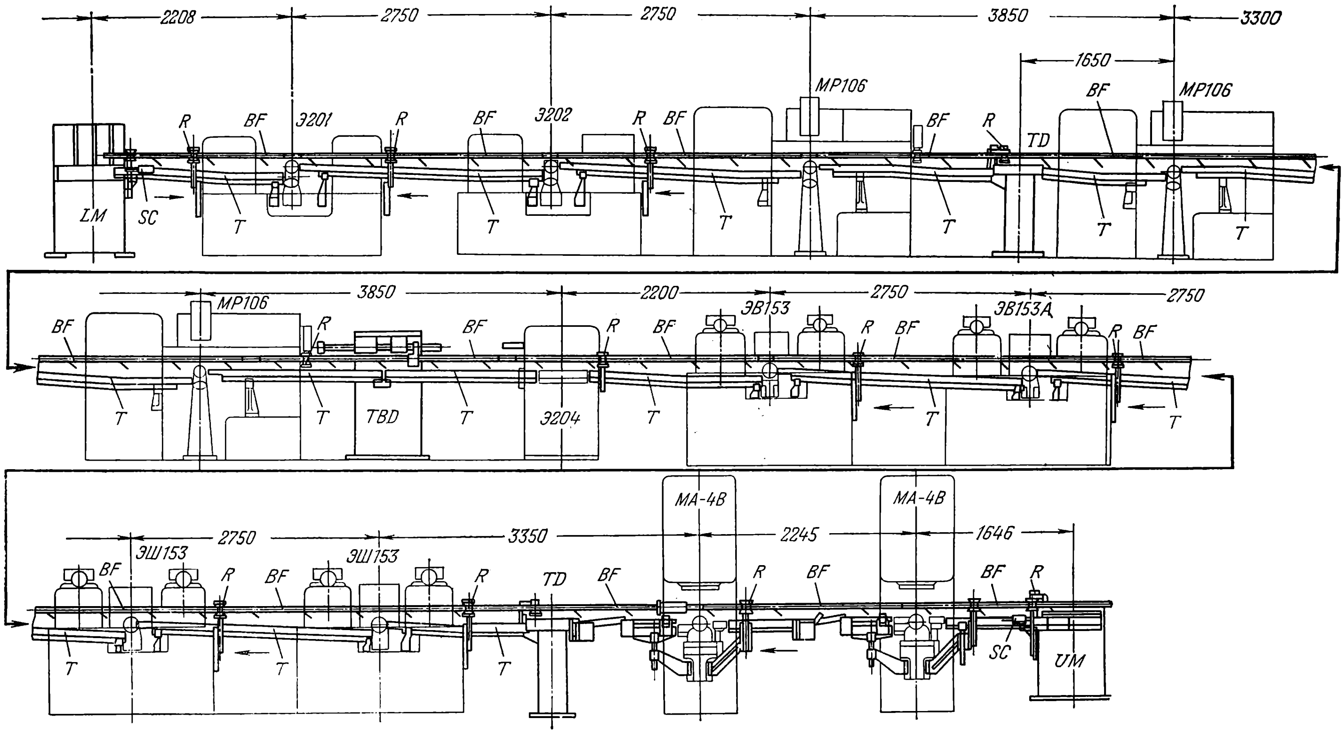


Fig. 264. General view of the handling system of the model Э-200 transfer machine:

LM—loading magazine; *SC*—counter of the passing shafts; *R*—roller support of the transfer bar; *T*—trough; *BF*—transfer bar with disappearing fingers; Э-201—shaft end face milling machine; Э-202—unit-built centring machine; *TD*—turning device; *TBD*—transfer bar drive; *UM*—unloading magazine

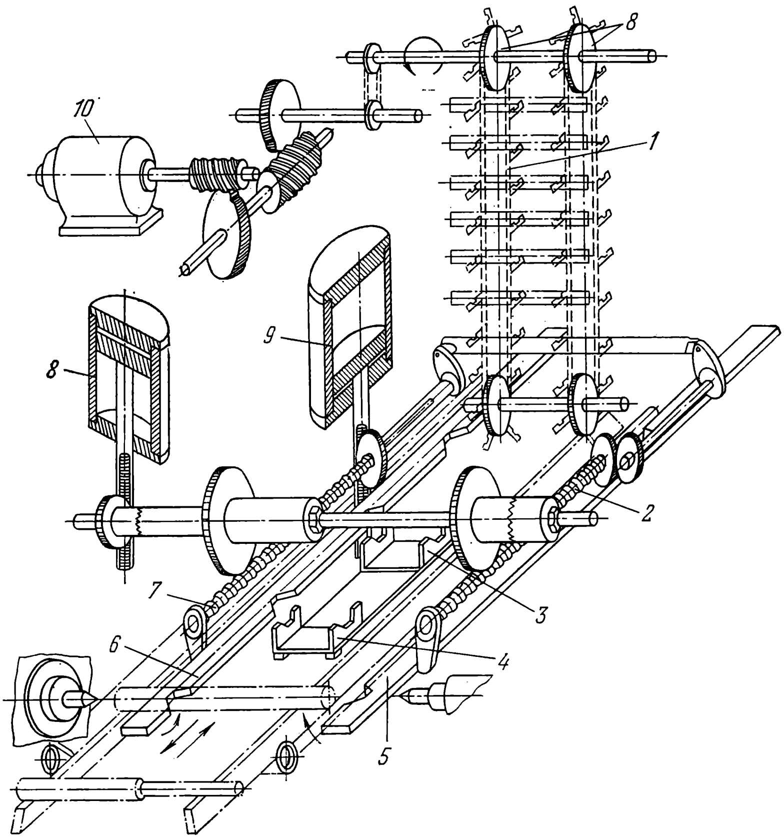


Fig. 265. Principle of the loading mechanism for the model MP-107 hydraulic tracer-controlled lathe

operates a limit switch. To reduce the stroke of the rotating transfer bars *5* and *6* and the time required for them to carry the blank from the chain-type magazine to the line of centres of the lathe, the distance from the magazine to the centre of the lathe has been divided into three parts. Two stationary V-blocks *3* and *4* are provided at the boundaries of the three parts. These V-blocks receive the blanks at the intermediate points during their transfer for loading. The parallel rotating bars *5* and *6*, turning about axles *2* and *7* by the action of hydraulic cylinder *9*, raise the next shaft blank, which is in the initial position in the chain-type magazine, as well as the blanks lying in the intermediate V-blocks *3* and *4*, and also receive the finished shaft

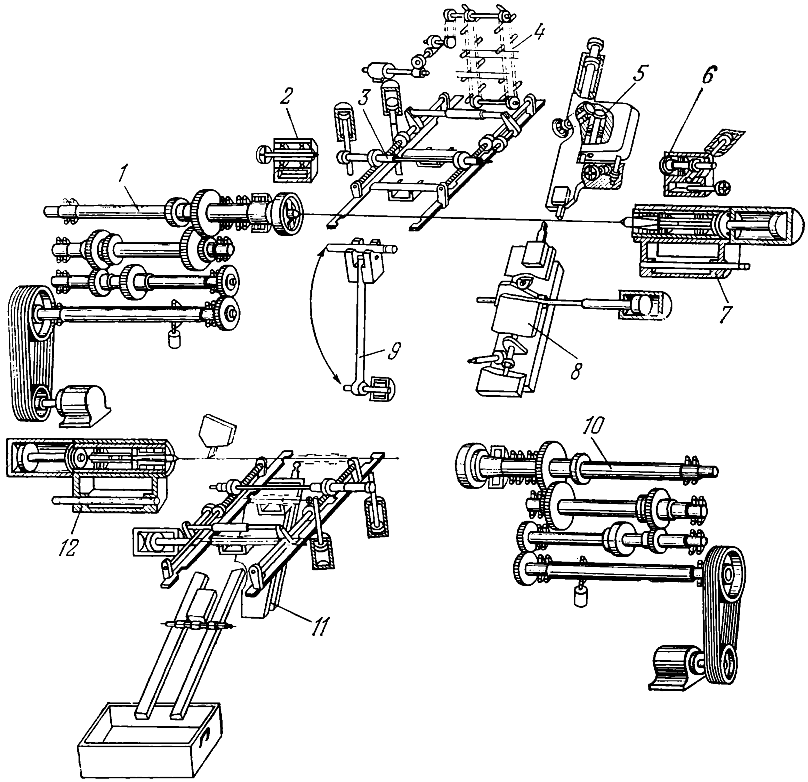


Fig. 266. Principle of the transfer machine, model MP-107:
1—spindle of lathe No. 1; *2* and *6*—template indexing mechanism; *3*—loading mechanism; *4*—chain-type magazine; *5*—tracer-controlled slide; *7* and *12*—tailstocks; *8*—facing slide; *9*—transloader; *10*—spindle of lathe No. 2; *11*—unloading mechanism

from between the lathe centres. All of these shafts (blanks and the finished workpiece) are held in V-slots of the transfer bars and are advanced in the forward stroke of the bars (actuated by hydraulic cylinder *8* through rack-and-pinion drives to axles *2* and *7* which constitute round racks) by the pitch of the transfer device which equals the distance between intermediate V-blocks *3* and *4*.

At this the finished workpiece is removed from the lathe, the next blank is delivered to the line of centres and two blanks are located above the intermediate V-blocks. As the transfer bars turn downward, they lower the workpieces into the intermediate V-blocks, between the lathe centres and onto

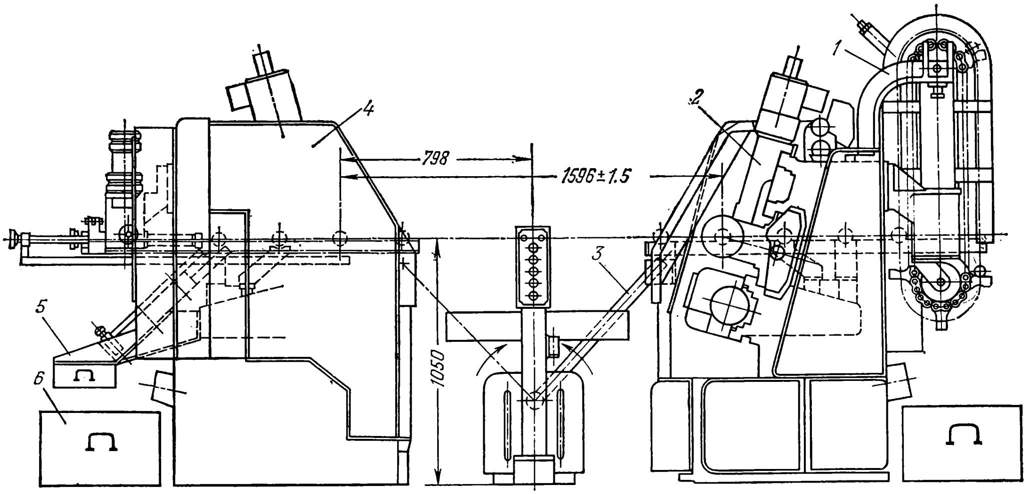


Fig. 267. Layout of the model MP-107 transfer machine:
1—chain-type magazine; *2*—first lathe; *3*—swinging transloader; *4*—second lathe; *5*—container (tote box) for the finished workpieces; *6*—chip box

the V-blocks of the transloader with spring grips (Fig. 266) which delivers the workpiece between the centres of the adjacent lathe.

A device, identical to the one described above, is used on the adjacent lathe to unload the finished shafts from the line of centres to a container. It takes 9 or 10 sec to unload a finished shaft and to load the next blank.

At the moment the blank is advanced by the loading transfer bars to the line of centres of the lathe, the tailstock spindle moves to its middle position in which the spindle centre enters the centre hole but does not shift the blank forward. This avoids breakage of the loading conveyer when parts of complex shape are being machined. The tailstock spindle is fixed in this position since both ends of its hydraulic cylinder are closed off by the valve on the hydraulic control panel.

The front (headstock) centre is advanced by its hydraulic cylinder up tight into the centre hole. Then the loading transfer bars turn downward and are retracted. After this the tailstock spindle forces the blank forward until its end face registers against the locating face on the driving chuck, thereby overcoming the force of the headstock centre whose hydraulic cylinder has a 65-mm bore while the bore of the tailstock spindle cylinder is 105 mm. The blank is secured by a self-clamping centrifugal driving chuck.

Weak-current contacts are provided in the V-blocks of the transloader to check whether they are holding a workpiece. If there is no workpiece in the

V-blocks, the transloader will not rock toward the second lathe which will not be started. In the vertical position of the transloader (Fig. 267), the lathes are accessible for servicing.

No stockpile bank has been provided between the lathes on the model MP-107 transfer machine, this being a shortcoming of this design. Upon the further development of transfer machines with through-horizontal transfer, five hydraulic tracer-controlled lathes, modifications of models 1712 and 1722, were used in the transfer machine, as well as a model MP71 milling and centring machine that allows through transfer. Vertical chain-type magazines were provided between the lathes for use as stockpile banks.

CHAPTER 20

TRANSFER MACHINES FOR GEAR PRODUCTION

Toothed gears are manufactured, both under mass production conditions in automatic machine tools which are rarely changed over, and in plants engaged in the lot production of a great number of types and sizes of gears.

Transfer machines for lot production are built to accommodate a group of gears of a definite type and in a definite size range. A standard manufacturing process is planned for this group (or family) of gears, based on the use of change-over-type transfer machines made up of general-purpose machine tools or their modifications.

Because of the high required output in the mass production of gears, transfer machines for this purpose are usually of the multiple-flow type. In this case the transfer machine is also made up of general-purpose machine tools, whose design has been perfected to a greater degree due to their wider application. Another reason for their use is that the transfer machine can be more easily changed over if product changes are made.

The choice of a handling system for a transfer machine to produce gears depends, not only on the size of the gears, but to an equal extent on the required output.

If the volume of production is comparatively low, only one machine tool will be required for each operation with the exception of gear hobbers, of which two or three may be needed. The transfer machine may be designed with a single-flow line served by a transfer bar. This bar is arranged for delivering blanks consecutively to the gear hobbers, skipping those that have not finished hobbing a preceding blank (transfer machine for single-rim gears). Transfer machines for a large volume of production are of the multiple-flow type and must be equipped with a distributing conveyer which can deliver a blank by the request of any of the machine tools operating in parallel.

In a transfer machine built in the Buick plant (USA) for the mass production of small gears (28 mm in diameter), trough conveyers with distributing baffles are used. The baffles are installed opposite vibratory-bowl feeders which supply each machine tool.

Fifteen gear shapers operate in parallel in the gear-cutting section. The other sections are also made up of multiple-flow lines. This feature, in con-

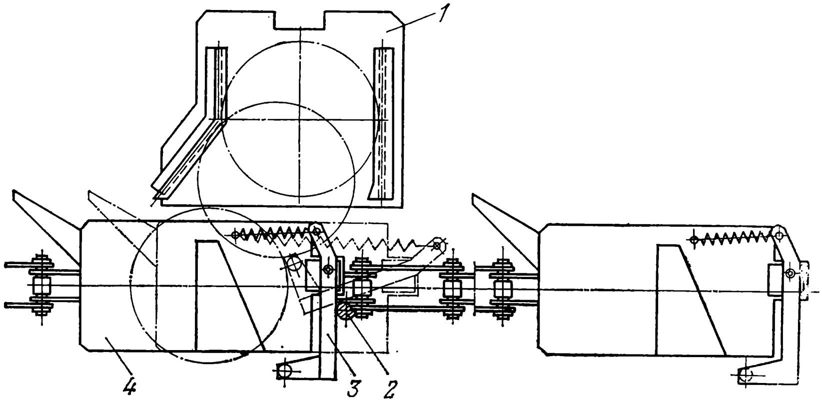


Fig. 268. Workpiece delivery from a conveyer truck to the input trough of a machine tool

junction with the use of hoppers at the component machine tools, substantially raises the utilization factor of the transfer machine.

Distributing conveyers of another type (due to the large size of the gears) have found application in a multiple-flow transfer machine which manufactures the large bevel gears in the rear axle unit of trucks.

A longitudinal endless single-chain truck conveyer is used for carrying the bevel gear blanks and distributing them among five semiautomatic eight-spindle vertical chucking machines operating in parallel. The blanks are carried in cells of the trucks. Blank delivery to the horizontal input trough of the machine is shown in Fig. 268.

If the input trough *1* is empty, stop *2*, arranged in front of the corresponding chucking machine, is raised by a solenoid. Upon motion of the conveyer truck *4*, lever *3* runs up against the stationary stop. This swings the lever to the side so that it pushes the gear blank into input trough *1* of the chucking machine. After this, stop *2* is lowered by the solenoid and subsequent blanks ride past the machine.

Transfer Machine for Single-Rim Spur and Helical Gear Production

The transfer machine described below was designed by the ENIMS Institute for manufacturing ten types of single-rim spur and helical gears (being changed over for each gear). It accommodates gears of the 2nd accuracy class (according to the USSR Std) with an outside diameter in the range from 88 to 220 mm and a module from 1.75 to 4 mm. These gears are com-

ponents of the model 1K62 engine lathe produced by the Krasny Proletary Plant and the required output is 120,000 pcs per year in lots of 1,000 pcs (Fig. 269).

It was planned to produce one type and size of gear in the transfer machine for five or six shifts. The transfer machine can be changed over by three setters-up in 4.5 hours.

The manufacturing process for machining the die-forged blanks with pierced holes is illustrated in Fig. 270. In-process inspection of the gears is carried out by the setters-up (operators) in instruments outside of the transfer machine.

The transfer machine is made up of single-station general-purpose machine tools of the vertical type (enabling the construction of the machine tools to be tested under various operating conditions, and to increase their dependability as a consequence) and of a new type (this could prove inexpedient if the construction had not been tested sufficiently in operation and suitably improved). One machine tool is sufficient for each operation, except for hobbing for which two machines are required, as can be seen in the cyclogram of transfer machine operation (Fig. 271).

The transfer machine has an in-line layout (see Fig. 269) with frontal workpiece handling by a single common transfer bar. Nevertheless, the transfer machine is divided into two sections by a stockpile bank which follows the finish turning operation. The workpieces are fed to the component machine tools from the transfer bar arrangement by transverse loaders.

The stroke of the transfer bar is 900 mm, but the fingers suspended from the bar advance the workpieces various distances, from 300 to 790 mm. Cams arranged at various points along the transfer bar raise the hanging fingers to by-pass certain workpieces. The workpieces are advanced a double pitch at the gear hobbers to divide the flow line.

The transfer machine is loaded by an automatic manipulator (Fig. 272) which removes the blank from the stud of a rotary magazine (Fig. 273) and delivers it to the transfer bar conveyer. The transfer machine is unloaded in the same manner by another manipulator which delivers the finished gear from the transfer bar, putting it on the stud of a rotary magazine. The self-clamping chuck *4* of the manipulator (Fig. 272) travels vertically with bracket *6* along column *5*, being actuated by a chain drive. This drive is powered by electric motor *1* through worm gearing, a bevel gear reversing unit and overload clutch *3*.

As spring-operated chuck *4* runs against the blank in its downward travel it grips the blank, taking it off the magazine stud. This occurs at the loading end of the transfer machine and at an odd number of applications of pressure under the chuck. At even applications of pressure, the chuck releases the blank, leaving it on the transfer bar conveyer. When pressure is applied under the chuck, overload clutch *3* is tripped, the chuck stops, and

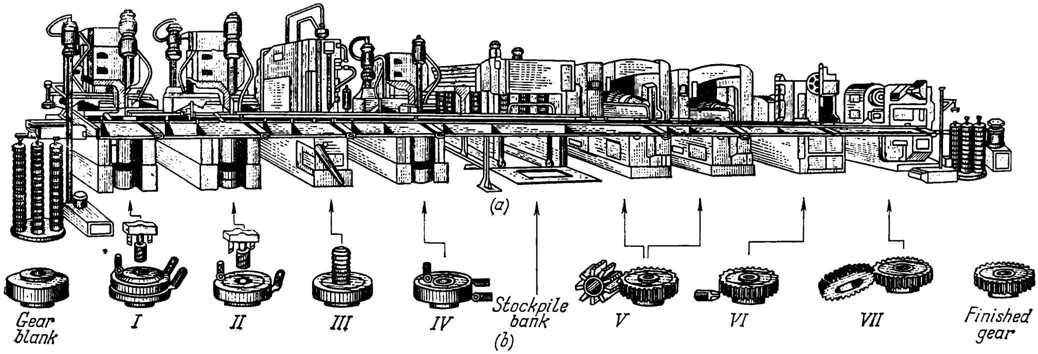


Fig. 269. Transfer machine for single-rim gear production:
(a) general view; (b) sequence of machining operations

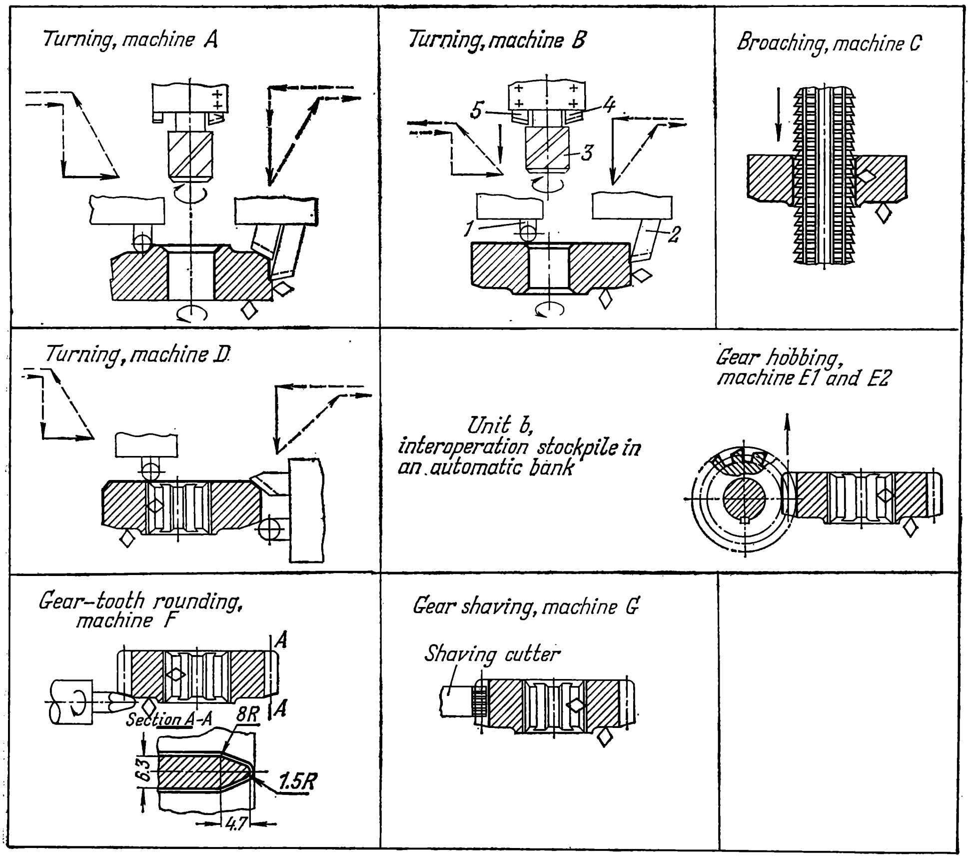


Fig. 270. Manufacturing process for machining single-rim gears

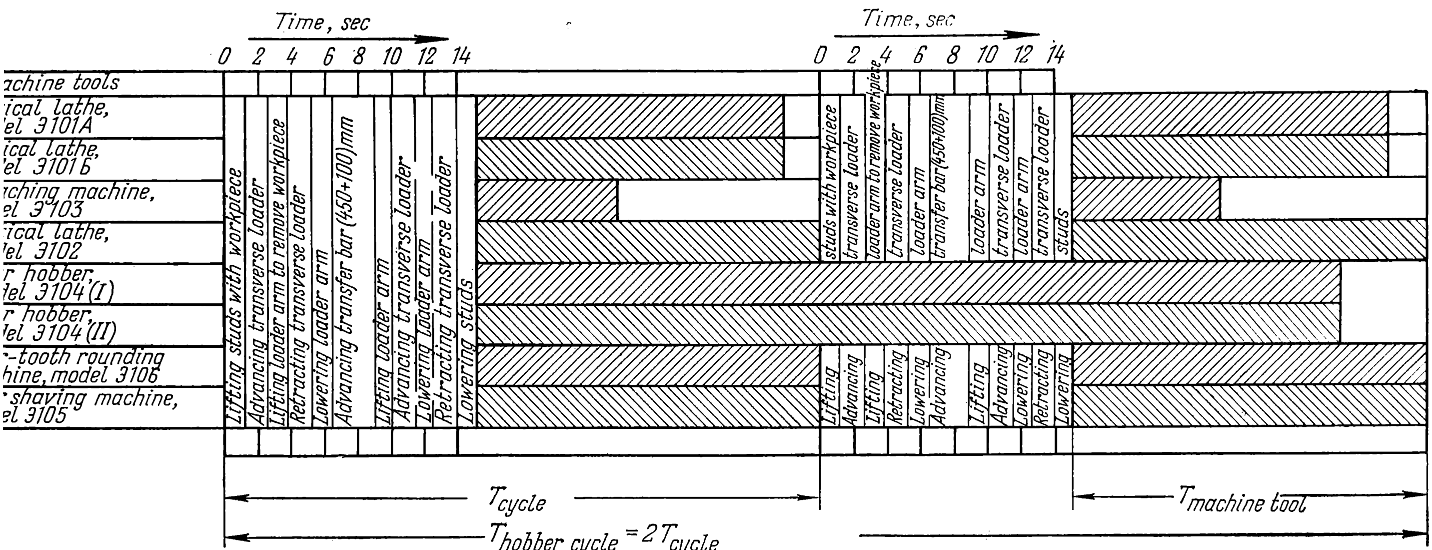


Fig. 271. Cyclogram of transfer machine operation in the manufacture of single-rim gears

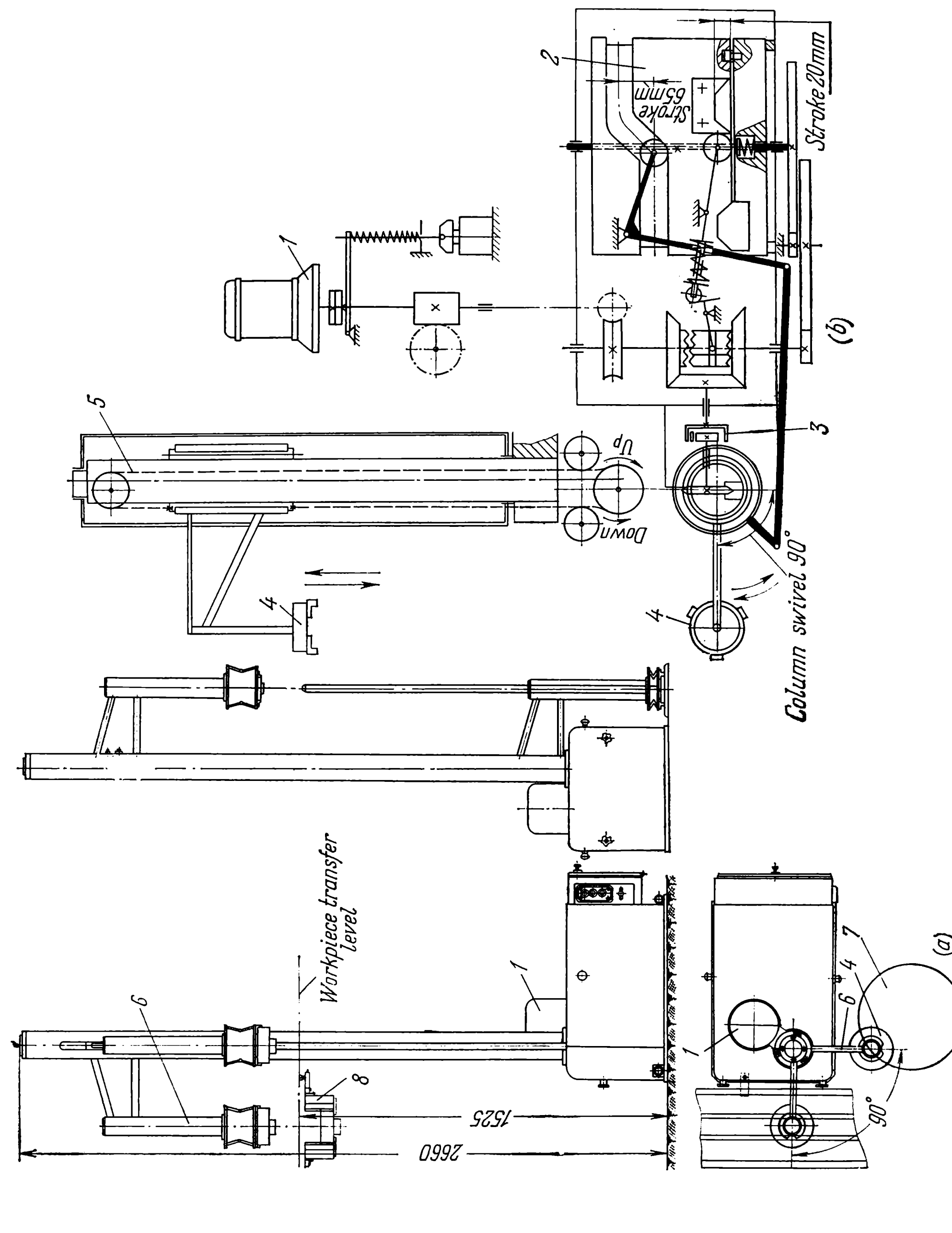


Fig. 272. Loading-unloading manipulator of the transfer machine for single-rim gear production: (a) general view; (b) gearing diagram

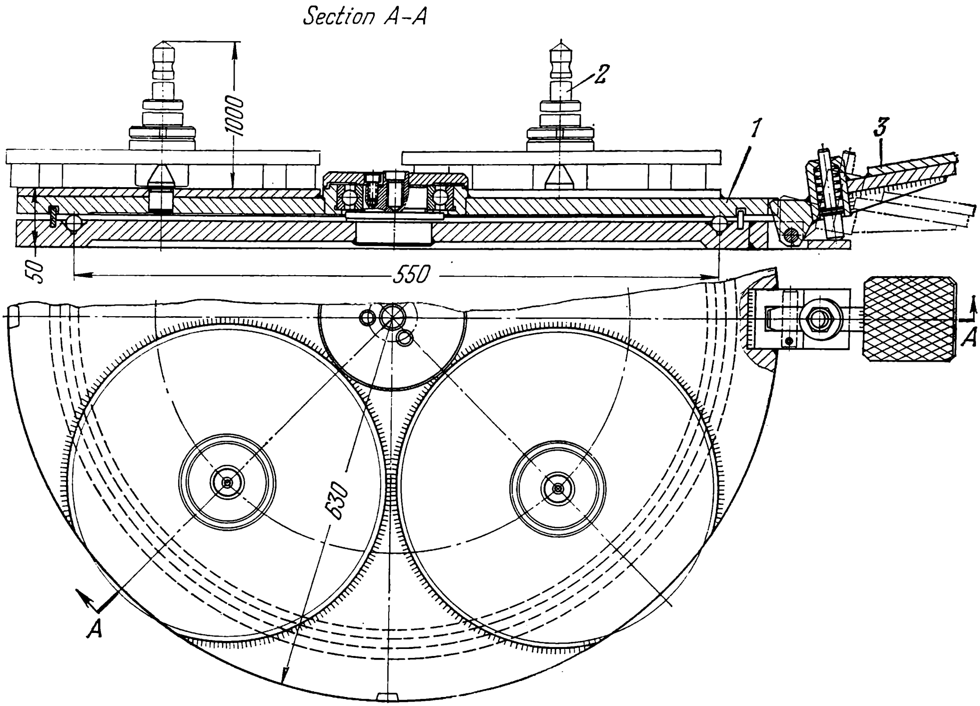
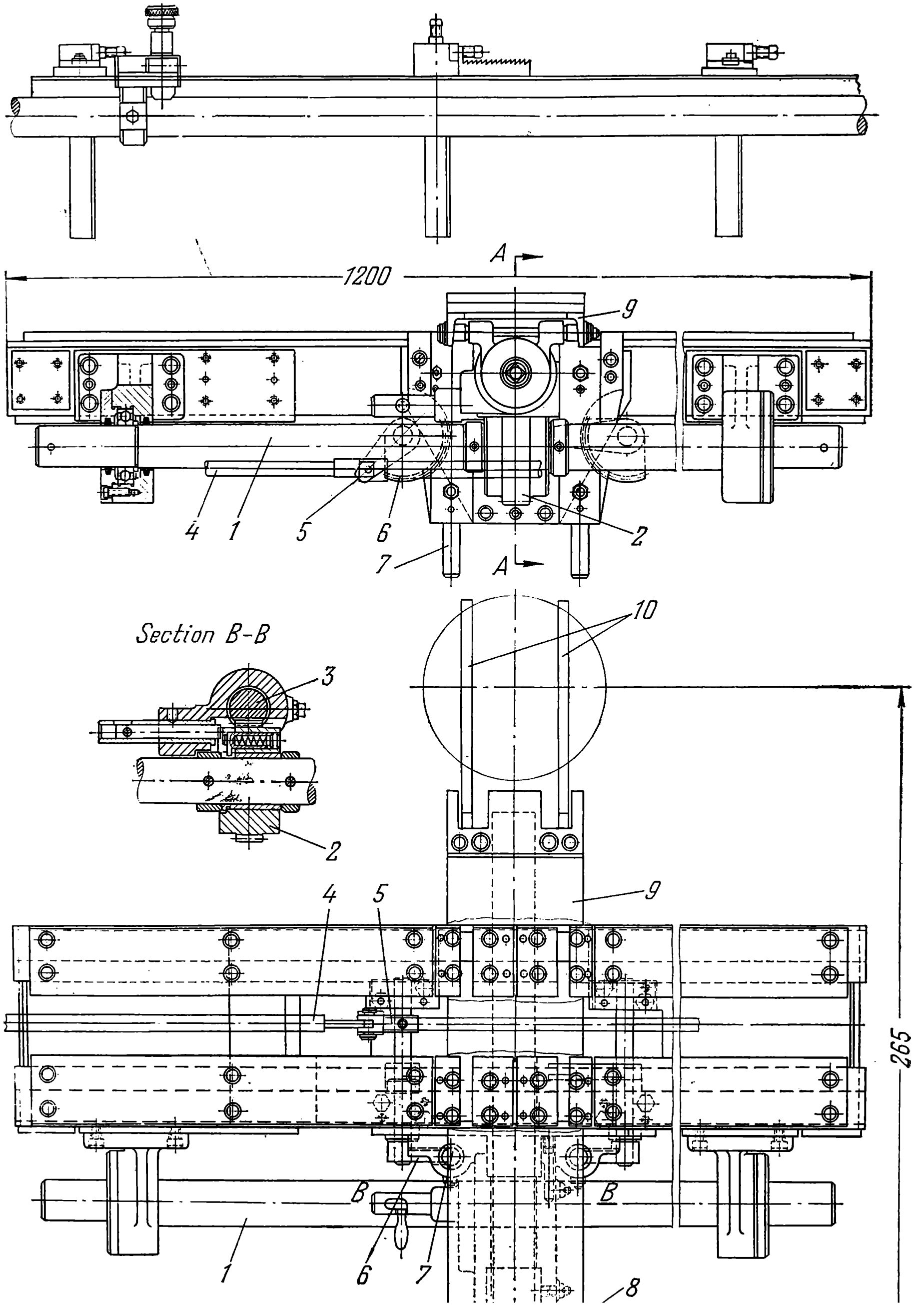


Fig. 273. Blank and workpiece magazine:
1—tray with vertical studs *2* for holding blanks or workpieces; *3*—pedal for indexing the tray

then the reversing unit, controlled by cam drum *2*, is operated. At this, the bracket with the chuck is raised until the lift limiter is reached. When overload clutch *3* is tripped, the reversing unit reverses motion and the cycle continues. Through a lever system, cam drum *2* swivels column *5* with bracket *6*, and the chuck lowers the blank (or workpiece at the unloading end) until it runs up against the conveyer or a tray or a finished gear on the stud of the unloading magazine. Then the chuck releases the blank or finished gear and is raised again. When it reaches its upper position the column turns back to the initial position.

When one of the studs of a magazine is empty or is filled to the top (at the unloading end), the operator releases the locking device of tray *7* of the magazine by depressing a pedal. He then indexes the tray so that the next stud is turned to the working position.



1200
A
A
9
4
1
5
6
2
7
10
Section B-B
3
2
9
4
5
265
B
B
1
6
7
8

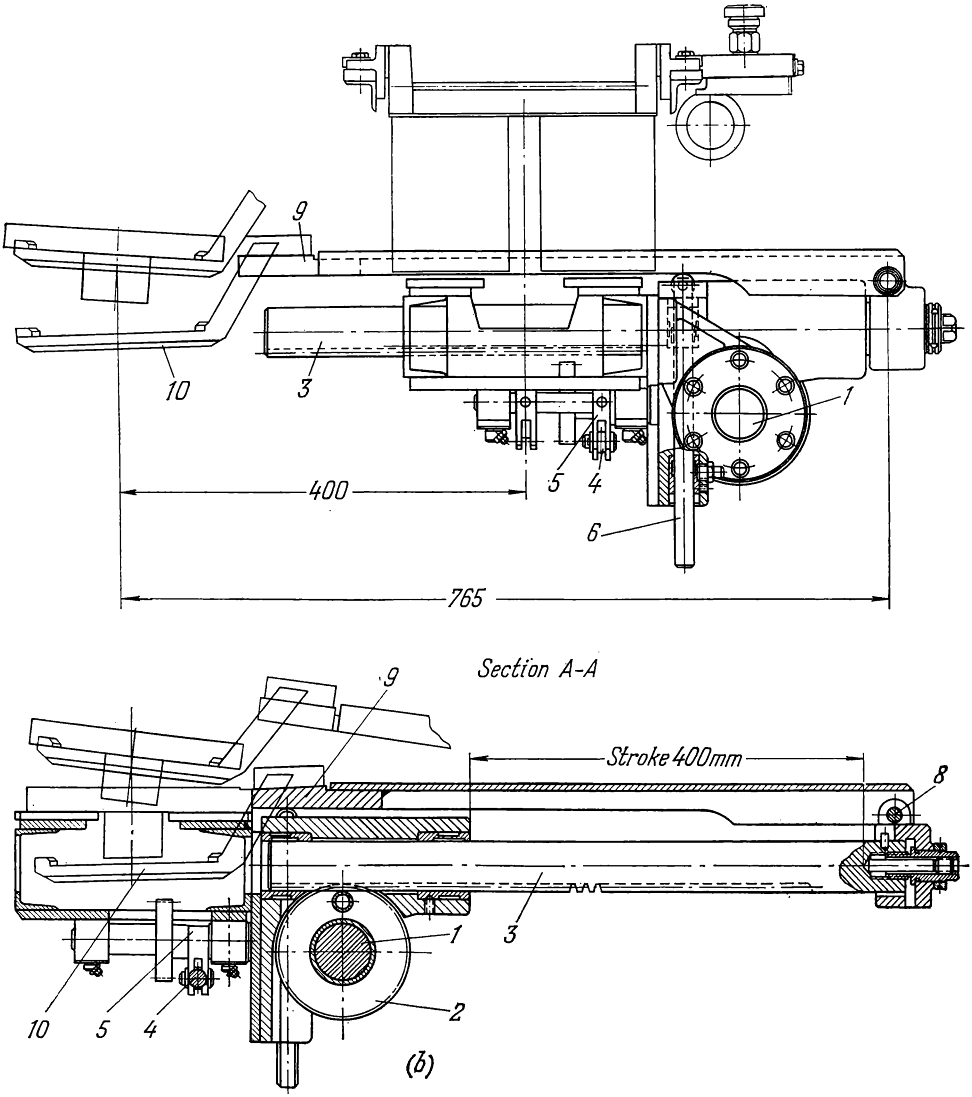


Fig. 274. Transverse loader:
(*a*) front and top views; (*b*) side view and longitudinal section

The transverse loader, in delivering the workpiece from the transfer bar conveyer to the machine tool, moves it in the cross horizontal direction, as well as vertically. The transverse loader (Fig. 274*a* and *b*) is used at all the stations. Its cross motion is obtained from shaft *1*, passing along the whole transfer machine, through pinion *2* and driving rack *3* to which the arm *9* of the loader is linked by pin *8*. Longitudinal motion of tie-rod *4* turns lever *5* and segment gears *6*. These segment gears mesh with round pushing racks *7*. The racks move upward, raising arm *9* and grips *10* by turning them about pin *8*. With machine tools in which the gear blanks are machined on spline arbors, special mechanisms are used for automatically mounting the gear blank on the arbor and other mechanisms for removing the machined gear from the arbor.

CHAPTER 21

CONTINUOUS ROTARY MACHINE TRANSFER LINES

21-1. Layout and Main Features

Rotary machine transfer lines are made up of rotary machine tools and rotary loading and handling devices (Fig. 275) which are in continuous rotation.

The workpieces are carried from machine to machine, the blanks are loaded and the finished workpieces are removed by means of rotary loading and handling devices. Machining is carried out in parallel at all the stations of a rotary machine tool to the same cycle but the cycle phases are shifted in respect to each other. The pace of a rotary machine transfer line and its output depend upon the number of stations in the rotary machine tools and the speed of rotation of their rotary tables.

The following features of rotary machine transfer lines distinguish them from other types of transfer machines:

1. The output does not depend directly upon the length of the operations since the output can be increased by increasing the number of stations on the rotary machine tools.

2. It is possible to obtain an equal output for all the component machine tools of the transfer line even though the length of the various operations may differ.

3. Though machining is done in parallel (with a phase shift) and each rotary machine tool has several stations, there is only a single common handling line, and not several parallel lines as in multiple-flow line transfer machines. This leads to more intensive utilization of the handling means—the rotary handling devices and the rotary tables of the component machine tools.

Owing to the possibility of obtaining equal output of the rotary machines by a proper selection of the number of spindles (stations) it is possible to combine operations in a single transfer line which could not be feasibly combined in a transfer machine made up of nonrotary machine tools. For example, in producing a small spindle, the workpiece is to be drilled, reamed and hardened. Let us assume that drilling and reaming require 10 min each, and hardening requires 2 min. Complex handling devices would be required to combine five nonrotary drilling machines and five nonrotary reaming machines with a single heat-treating installation. On the other hand, two rotary machine tools with 20 stations each for drilling and reaming, a single

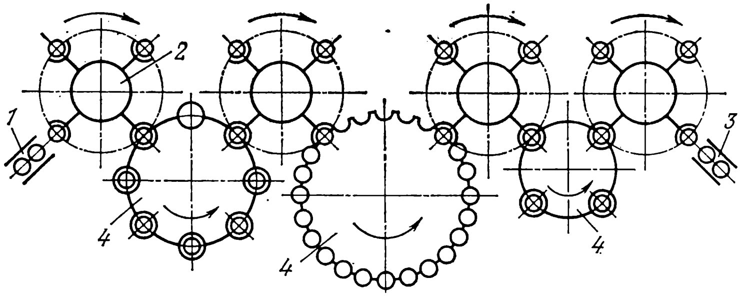


Fig. 275. Layout of a rotary machine transfer line:
1—loading chute; *2*—rotary handling device; *3*—delivery chute; *4*—rotary machine tool

four-station rotary hardening device and two rotary handling devices make up a transfer line with an output of 120 pcs per hour (Fig. 276).

A continuous rotary machine tool consists of a group of toolholders *8* (Fig. 277), arranged in a circle on a common shaft about which they rotate continuously. During this rotation each tool is reciprocated by the upper and lower slide blocks, *4* and *10*, according to a cycle dictated by the profiles of the upper and lower stationary cams, *5* and *11*. The workpieces are loaded and unloaded by rotary handling devices *12* and *13* which are equipped with grips *7* and *9*. The rotary handling devices are kinematically linked to shaft *6* of the rotary machine tool through gears *1*, *2* and *3*.

The layout of the rotary machine tools is mainly based on the use of centrally located cutting tools (drills, reamers, taps, etc.) operating on simple linear cycles (i.e., without transverse movements). This limits the appli-

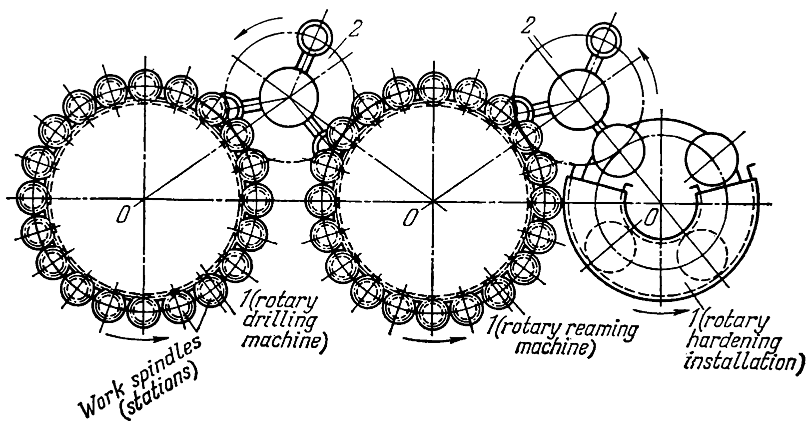


Fig. 276. Combining drilling, reaming and hardening in a single rotary machine transfer line

cations of rotary machine tools for metal-working production.

Rotary machine transfer lines are of the interlocked type. Malfunctioning of any element leads to downtime of the whole transfer line. In these transfer lines a high production capacity can be ensured with operation at relatively low cutting speeds and a long tool life. This enables downtime for changing and adjusting tools to be reduced. Devices for automatic inspection and for automatic tool changing can be built into rotary machine transfer lines.

If the feed drive is of the mechanical type, the motion of the workpiece in the rotary machine tool coincides with the drive of the working members from stationary cams. This simplifies the rotary motion drive, the feed drive and the control of their cycle of motion. This feature, in turn, considerably increases the dependability of rotary machine transfer lines in comparison with ordinary interlocked transfer machines.

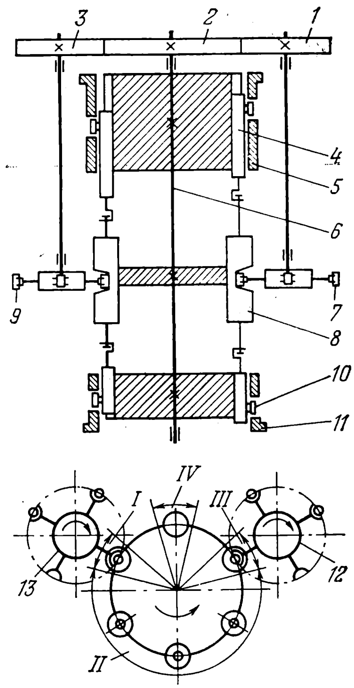


Fig. 277. Principle of a rotary machine tool

21-2. Output of Rotary Machine Transfer Lines

The output of a rotary machine transfer line is

$$Q_r = \frac{M}{t} \text{ pcs per min} \tag{102}$$

where M = number of spindles (stations) in a rotary or nonrotary machine tool

t = time required to machine one workpiece, min

$$t = t_m + t_h \tag{103}$$

where t_m = machining (processing) time, min

t_h = handling time, min.

The speed of the rotary table is

$$n = \frac{1}{t_m + t_h} \text{ rpm} \tag{104}$$

The output of a nonrotary transfer machine or line is

$$Q_n = \frac{M}{t_m + t_{hn}} \text{ pcs per min} \tag{105}$$

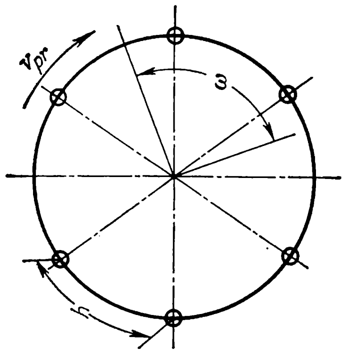


Fig. 278. Diagram of work spindle arrangement in a rotary machine tool

while that of a rotary machine transfer line is

$$Q_r = \frac{M}{t_m + t_{hr}} \text{ pcs per min} \tag{106}$$

Since each spindle (station) delivers one finished workpiece as it passes through the unloading zone, the output equals the number of spindles passing through this zone in a unit of time. Thus

$$Q_r = \frac{v_{pr}}{h} \text{ pcs per min} \tag{107}$$

where v_{pr} is the peripheral velocity of the centres of the work spindles

$$v_{pr} = \pi D n \text{ m per min} \tag{108}$$

where h is the arc of the circle in metres between the axes of the spindles (Fig. 278).

The workpieces are loaded and unloaded during the time t_{hr} min as the rotary table turns through the angle $\omega°$.

Consequently

$$\frac{t_{hr}}{t_m} = \frac{\omega}{360 - \omega} = k_r \tag{109}$$

or

$$t_{hr} = k_r t_m \tag{110}$$

Substituting into equation (106), we obtain

$$Q_r = \frac{M}{(1 + k_r)\, t_m} \tag{111}$$

A comparison of curves Q_n and Q_r, plotted against the machining (processing) time t_m, is given in Fig. 279.

With an increase in t_m, the output Q_r drops at a faster rate than Q_n because t_{hr} depends upon t_m while t_{hn} does not. From the equation $Q_n = Q_r$ it follows that $t'_m = \frac{t_{hn}}{k}$. At $t_m < t'_m$, theoretically $Q_r > Q_n$.

The first rotary handling device is to be loaded with M workpieces during the time $t_{rhd} = t_m + t_{hr}$. Hence

$$t_{rhd} \geqslant M t_{lr} \tag{112}$$

where t_{lr} is the time required to load one workpiece. This limits the minimum value of t_m to a certain value t''_m (Fig. 279). Theoretically, at a processing time t_m, beyond the limits t''_m and t'_m, the output of rotary machine

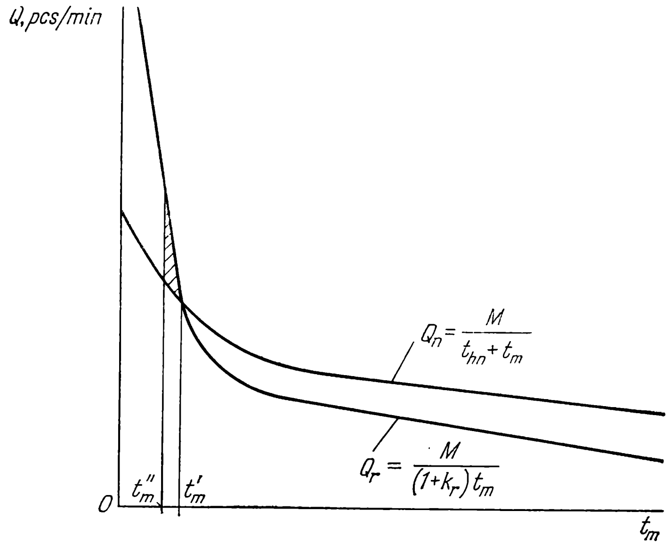


Fig. 279. Comparison of the outputs of nonrotary transfer machines and rotary machine transfer lines

transfer lines is lower than that of corresponding nonrotary transfer machines. The field of efficient application of rotary machine transfer lines is workpiece machining with a short processing time t_m. If the operation of M spindles (stations) in parallel (with a phase shift) is taken into account, the output pace should be a fraction of the machining time, i.e., very short. This is possible in the mass production of small simple parts, if the accuracy requirements are such that they permit the workpiece to be reclamped several times in its manufacture.

Rotary machine transfer lines have found application primarily for the mass production of small parts requiring a simple manufacturing process.

CHAPTER 22

TRANSFER MACHINE CONTROL SYSTEMS

The majority of the component units and mechanisms of a transfer machine are equipped with separate drives and constitute fully automatic units or sets of units, each with its own control system (usually of the in-travel type) providing for its particular automatic or semiautomatic working cycle.

When a machine tool is built into a transfer machine, its automatic machining cycle should be co-ordinated with the working cycles of the handling and loading systems. To this end, commands should be incorporated in the control system for starting the machine tools at the end of the handling and clamping cycle, as well as a command for starting the transfer system at the end of the machining cycle.

Along with these commands, an integrated system of automatic controls provides for the automatic action of the transfer machine. This system includes:

(1) a system for controlling the sequence of action of the component units of the transfer machine;

(2) an interlocking system ensuring trouble-free operation of the transfer machine;

(3) an adjusting system for setting up and readjusting the machine tools and cutting tools so as to obtain stable workpiece dimensions;

(4) a system of checking the workpiece dimensions;

(5) a system of setting-up (manual) controls of units and cutting tools;

(6) a signalization system, facilitating transfer machine servicing, troubleshooting and the determination of the time for changing and readjusting the cutting tools.

The control system of a transfer machine should provide for two and sometimes three modes of operation:

1. Automatic, continuously repeated, working cycle.

2. Semiautomatic working cycle, in which the control system provides for automatic operation only within a single cycle; as the workpiece is delivered to the loading position, the operator must press the preliminary start push button. A semiautomatic mode of operation is used in very complex transfer machines.

3. Setting-up or manual operation, providing for individual start of the whole cycle and for stopping certain working units of the transfer machine. This feature is obtained by the provision of rotary setting-up switches by means of which any unit or group of units can be switched off in setting up.

In an integrated control system for a transfer machine, the operation of the independent sections and parts of the transfer machine is always co-ordinated by means of an electrical control circuit.

The operation of the separate component units in the sections of a transfer machine is co-ordinated with the general automatic cycle mainly by combined means of control: electromechanical, electrohydraulic and electropneumatic.

Depending on the type of operative mechanisms provided for the working members, various types of internal interlinkages of the automatic cycle controls are used for the various component units. They may be combined mechanical, hydraulic and pneumatic systems. The application of in-travel controls excludes the possibility of disadjustment of the particular cycles of the various working members of the units in space. It is preferable to start the various units of a section of the transfer machine by an electrical control circuit. The role of such circuits as a means of cyclic interlinkage between the units increases with the complexity of the transfer machine section.

22-1. Systems for Controlling the Sequence of Action of the Component Units of a Transfer Machine

Centralized, decentralized and combined systems are employed to control the sequence of action of the component units.

The choice of a control system depends upon the purpose of the transfer machine or line, its layout, dimensions, length of the cycle and mainly on the type of component equipment.

In particular cases, the operation of various units can be efficiently co-ordinated by introducing simple mechanical linkages between them (see Fig. 222), thereby increasing the utilization factor of the transfer machine. In general, the system for controlling the sequence of action of the equipment should be co-ordinated with the type of drive of the various units, their internal cyclic interlinkages and the control system for their specific automatic cycles.

Most extensively applied in the component units of transfer machines are drives with noncyclic operative mechanisms—hydraulic, pneumatic and mechanical drives with screw and nut, rack or chain mechanisms. The specific cycles of working members with noncyclic operative mechanisms are controlled by in-travel systems. Consequently such in-travel systems are

widely employed to control the sequence of action of the component units of transfer machines.

In-travel systems for controlling the specific cycles of the working members of the various units enable centralized action-sequence control of the units to be applied, without fear of disadjustment of the cycles in space, by signals transmitted from a continuously and uniformly rotating master control switch. However, due to the nonrigid kinematic characteristics of hydraulic drives, widely employed in transfer machines, the command signals should be transmitted according to the maximum possible length of the particular cycles. This leads to additional intracycle losses.

The master control switch consists of one or several disks carrying brushes which close contacts, or cam dogs, operating electrical limit switches or hydraulic or pneumatic valves located around the disks.

Such systems for controlling the sequence of the phases of the cycle are used in many transfer machines. Decentralized control systems with in-travel signals from various kinds of pickups are extensively employed because of their high flexibility and simple operation. When these systems are in proper order, they exclude the possibility of accidental engagements. Such systems, however, are not without shortcomings. The pickups for transmitting the command signals must be located, in many cases, in the working zones of the component machine tools and other units where they are subjected to the action of the vapours of the lubricants and cutting fluids, as well as metal dust. This may lead to malfunctioning and the transmission of false commands when circuits are shorted and be the cause of a breakdown.

A shortcoming of a decentralized control system is the large amount of electrical equipment required. This increases downtime due to malfunctioning of the control system.

Centralized systems with in-travel controls are more efficient. The shaft of the master switch is turned periodically by the signals from pickups which check the execution of previous commands along the line of travel of the working members, resistance to travel, rate of travel and other factors.

The shaft of the master control switch, carrying the disks with the command cam dogs, turns through the angle between consecutively located cam dogs, thereby switching over the control circuit for executing the subsequent phases of operations of the component units. The master switch shaft makes one revolution during one cycle of operation of the transfer machine.

Such a control system is very flexible. It enables the length of time allotted to the execution of each phase to be set up according to the time actually required, regardless of the lengths of the other phases. It also enables the execution of the various phases of the cycle to be checked, not only indirectly, but by direct indications of the changes in the dimensions and shape of the workpiece.

This feature is the main advantage of this system over centralized control systems in which the command signals are transmitted by a uniformly rotating master switch after definite intervals of time.

In comparison with a decentralized system, a master switch with periodic shaft rotation, being itself an interlocking device, enables the controlling functions to be performed with a fewer number of interlocking devices. Change-over to the setting-up (manually controlled) mode of operation is simplified. In setting up, the master switch shaft is turned to a new position by merely pressing a push button.

In this system of control, as in a decentralized system with in-travel control, the response time of the control system does not coincide with, or overlap, the time for executing the commands. This leads to additional time losses for the operation of the control system.

22-2. Means for Controlling the Sequence of Phases in Transfer Machine Operation

The particular automatic cycles of the machine tools and other units of a transfer machine are controlled by means based on electrical, hydraulic and pneumatic automation. The commands for the co-ordination of the operation of machine tools and other units with the general automatic cycle of the transfer machine are usually transmitted and their execution is checked by electrical automation devices, while means based on hydraulic and pneumatic automation are used to execute the commands.

A command pulse is produced by a *signal pickup*. It is amplified or transmitted farther with a certain delay in a *signal converter* and is realized in the *operative member*—the drive of the mechanism which accomplishes the given control action. In addition to devices of these three kinds, control means include master switches.

Depending upon the method of initiating the signals, signal pickups may be classified as in-travel, dimensional, force, power and velocity pickups. Most commonly applied in transfer machines are in-travel pickups in the form of limit switches, which transmit a signal indicating that a definite displacement has been executed. Pickups of the other types mentioned above are used to control the particular automatic cycles of the working members in transfer machines.

Ordinary limit switches are employed for speeds of travel over 0.4 m per min. At lower speeds, snap-action limit switches, usually of the two-way type, have found application. Limit switches may frequently become clogged with metallic dust, lubricant or cutting fluid, since they are commonly installed in or near the working zones of the machine tools or other processing units. This reduces their dependability; they sometimes fail to operate or

are shorted. For this reason, contactless limit switches are being employed more and more in recent years. They are distinguished for their high accuracy of operation, high dependability at a wide range of travel speeds of the actuating cam dog and practically unlimited service life.

Auxiliary and time-delay relays are employed to convert the command pulses of the signal pickups.

Auxiliary relays are used to amplify the signal pulse by closing a circuit supplied by current of higher power. They are used as a supplement to a limit switch when the latter lacks sufficient contacts to transmit the required signals.

The transmission of a command is most often delayed by means of a pneumatic time-delay relay.

Squirrel-cage induction motors are employed as electrical operative means for controlling the drives of the component units of a transfer machine. Frequently these motors are equipped with built-in brakes. Electromagnetic clutches and solenoids for actuating valves are also employed as operative means.

Shown in Fig. 280 is the circuit diagram of a master switch in which the shaft has periodic angular rotation actuated by signals indicating that preceding commands have been executed.

Mounted on shaft *5* of the master switch are cams *6* which, through lever system *9*, shift the control valves *8* of the hydraulic system of the transfer machine. These cams ensure the proper sequence of phases of the transfer machine cycle during one revolution of the master switch shaft. Mounted on, but electrically insulated from, shaft *5* are commutator slide *2* and disk *1*. Shaft *5* is turned by ratchet wheel *10* upon oscillation of segment gear *3* with pawl *4* about the shaft. Piston rod *16* on which rack teeth are cut rocks the segment gear. The rod also carries double stop *12* which closes limit switch *11* during the up stroke of the piston rod and opens it on the down stroke.

Arranged on disk *1* are contacts *18*, each of which is connected to one branch of the supply circuit of limit switches with normally open contacts located along the line of travel. The second, common branch of these circuits is connected to the commutator disk through the coil of solenoid *15* of valve *14*, limit switch *11* and brush *17*. When the slide of the commutator is at any one of contacts *18*, solenoid *15* will remain de-energized until the next limit switch in the order of the operating cycle is closed, thereby verifying the execution of the preceding command. When the contacts of the limit switches along the line of travel are open, piston rod *16* is in its upper position and the contacts of limit switch *11* are closed.

The execution of the preceding command operates the corresponding limit switch along the line of travel; its contacts are closed, solenoid *15* is energized, valve *14* is shifted downward, working fluid is admitted to the

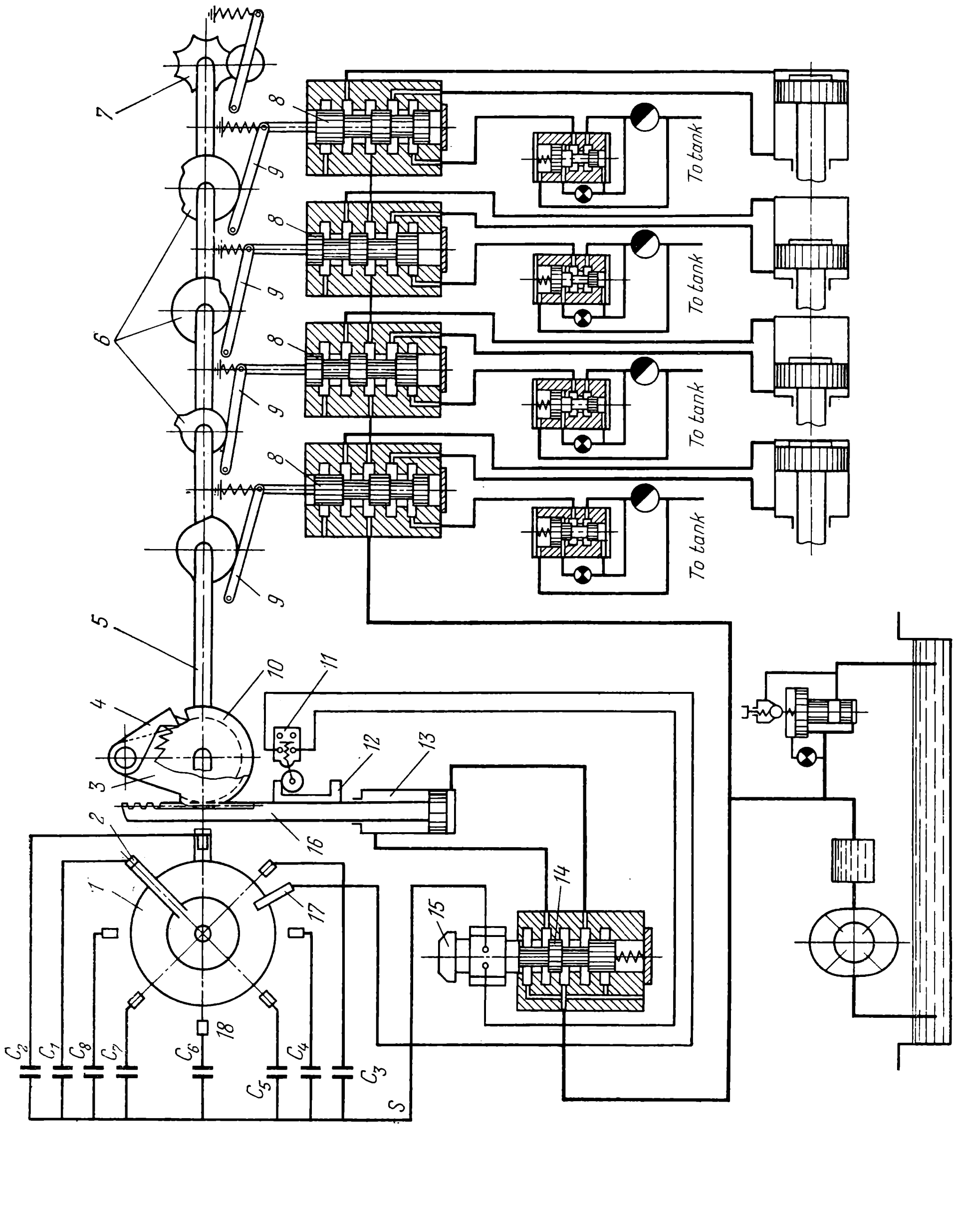


Fig. 280. Circuit of an electrical master switch designed for periodic rotation

upper (rod) end of hydraulic cylinder *13*, piston-rod-and-rack *16* travels downward and prepares pawl *4* for turning ratchet wheel *10*.

In the final lower position of piston rod *16*, double stop *12* opens the contacts of limit switch *11*, solenoid *15* is de-energized, valve *14* returns to the upper position, working fluid enters the head end of cylinder *13*, and piston rod *16* travels upward, turning shaft *5* through a definite angle by means of the ratchet and pawl mechanism. Sprocket *7* fixes shaft *5* in the indexed position.

Upon the rotation of shaft *5*, cam *6*, through a system of levers *9*, shifts control valves *8* of the transfer machine hydraulic system to the position corresponding to the current command, and slide *2* of the commutator stops at the next one of contacts *18*, preparing the succeeding command. Though limit switch *11* is closed at the upper position of double stop *12*, solenoid *15* remains de-energized until the limit switch, checking the execution of the current command, is closed. Then the operating cycle of the master switch is repeated.

22-3. Interlocking in Transfer Machine Control Systems

The system of interlocking switches off the component units of the transfer machine in which malfunctioning occurs or does not permit them to be switched on. In this way, the system ensures almost trouble-free operation of the machine tools, fixtures, cutting tools and other equipment.

Interlocking is commonly employed to check whether workpieces are properly located in fixtures, whether cutting tools are in proper order, and whether the lubricating, coolant and chip disposal systems are operating properly.

The proper position of a housing-type part in the working stations of a transfer machine is determined by the orienting, locating and clamping elements of the fixture. If the workpiece has not been correctly delivered into the fixture, the locating pins may enter a recess in the workpiece, instead of the locating holes. Then, at the end of pin travel, the system for controlling the sequence of the phases in the operation of the units will transmit a command for clamping the workpiece in the incorrect position.

To avoid the possibility of a breakdown, the position of a housing-type part is additionally checked before it is fixed by the locating pins by means of an electrical interlocking switch (Fig. 281) installed along the line of workpiece travel. A pneumatic pickup may perform the same function, being operated by the air pressure developed between the locating datum surface of the workpiece and the mating surface in the fixture (Fig. 282). If the workpiece is not properly located, not all of the holes will be closed

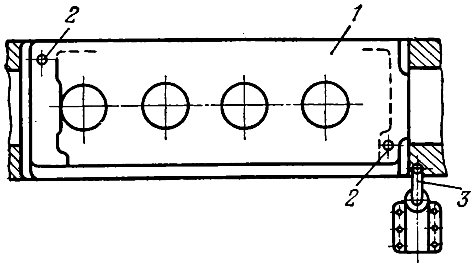


Fig. 281. Arrangement for tentatively checking the position of a cylinder block before it is fixed by the locating pins:

1—cylinder block; 2—locating datum holes; 3—interlocking switch

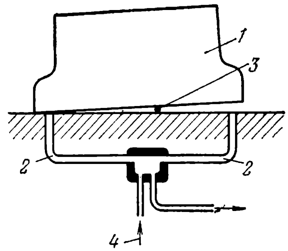


Fig. 282. Checking for proper location of a cylinder block by means of a pneumatic pickup:

1—cylinder block; 2—passages for delivering compressed air to the locating datum surfaces; 3—piece of chip on the locating surface; 4—compressed air supply from the mains to the pneumatic pickup

and the pneumatic pickup will transmit a command for stopping the transfer machine.

Sometimes an interlocking control device is installed between the stations at which holes are drilled and at which thread is tapped in these holes. This device (Fig. 283) checks whether the holes have been drilled and whether they are of sufficient depth, thus avoiding tap breakage. The hole depth is checked by entering probes into the holes. The probes are 1 or 2 mm less

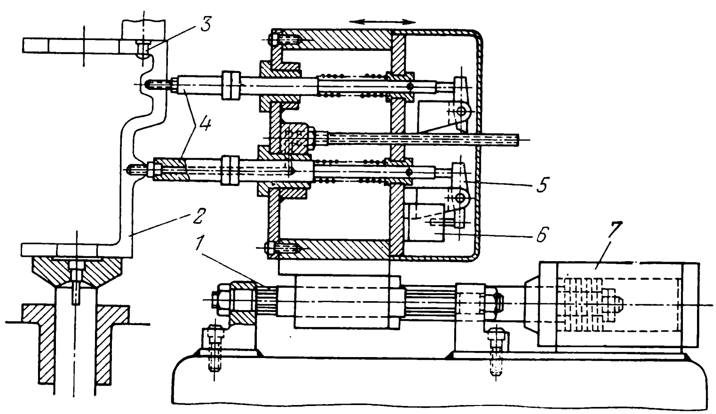


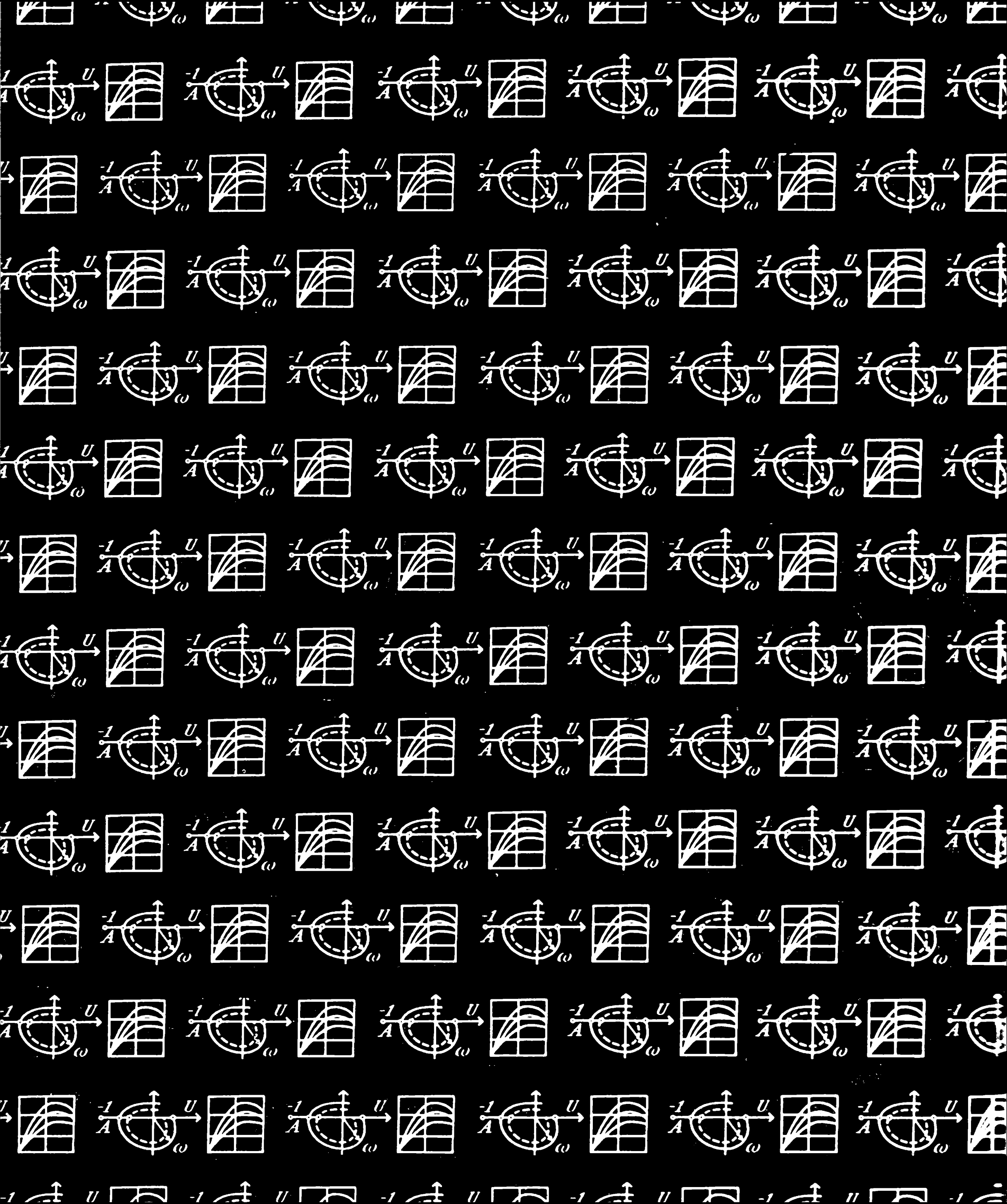
Fig. 283. Control station for checking the depth of drilled holes:

1—ways; 2—workpiece; 3—locating pin; 4—rod carrying the probe; 5—lever; 6—switch; 7—hydraulic cylinder

in diameter than the holes and are set to the required drilling depth. If the hole is too shallow or if a piece of a broken drill has jammed in the hole, the probe will be pushed back. This motion is registered by a microswitch, the corresponding section of the transfer machine is stopped, and a light signal indicates the group of tools in which the breakage has occurred. To avoid tap breakage in horizontal holes, the checking of their depth is combined with the blowing out of chips left by the drill. For this purpose, compressed air is delivered to channels provided in the probes. The application of a hole depth control device is shown in Fig. 218 (item *12*).

PART SEVEN

MACHINE TOOL TESTING AND RESEARCH



CHAPTER 23
MACHINE TOOL TESTS

Machine tools and their component units may be the objects of experimental research and are subject to various tests, differing in purpose and scope, in the course of the development of a new design, after manufacturing the pilot model, and following repairs and overhauls.

The requirements made to the accuracy of up-to-date machine tools and to their output, dependability and automated controls have greatly widened the range of tests and research, and have made them much more diversified, especially in respect to pilot models and prototypes.

The main types of machine tool tests conducted in the USSR are acceptance tests of lot-produced models and comprehensive tests of pilot models.

In addition to tests carried out under laboratory conditions, more fundamental and diverse research is being conducted on machine tools as a whole and on their constituent units and mechanisms. For this purpose special testing facilities are used which imitate the operation of some mechanism of the machine under various conditions and are equipped with the corresponding measuring apparatus. Extensive theoretical research, employing electronic computers and simulators, is being conducted at the present time in conjunction with the experiments.

Acceptance tests conclude the process of designing and manufacturing a lot-produced machine tool. These tests are simpler than all the rest (see below) and are usually conducted by the inspectors of the plant inspection department under shop conditions. Their purpose is to check the performance of the machine tool and whether it complies with the manufacturing specifications.

According to present-day manufacturing specifications, acceptance tests include preliminary trials and an idle-run test, performance tests under load, checking whether the machine tool complies with the accuracy standards, and tests to determine the rigidity and vibration-proof properties in metal cutting.

As a rule these tests give a general idea of the quality of the machine tool without taking much time or requiring complex apparatus. Therefore, at the manufacturing plant and following overhauls and even medium repairs, acceptance tests are to be conducted without fail. For this purpose the machine

tool is installed on a special foundation. With the aid of adjusting wedges or levelling shoes, positioned in the same manner as for installing the machine for regular operation, it is levelled in the longitudinal and transverse directions to a precise spirit level. The foundation bolts are not tightened. After a preliminary trial in which the various mechanisms of the machine tool are switched on to make sure that they are in proper order, the idle-run tests are begun. These tests commence with consecutive changes in the main drive speeds from the minimum to the maximum values. The machine should continue to operate at the maximum speed until the spindle bearings reach a steady temperature. This temperature should not exceed 85°C for rolling and 70°C for sliding friction bearings. In the other mechanisms, the bearing temperature should not exceed 50°C. At the same time, the operation of the feed mechanism is checked at the low, medium and high working feeds, as well as rapid traverse motions. The operation of all automatic devices, stops and indexing mechanisms is checked; backlash is determined for all the actuating screws of the manual controls; the operation is checked of the mechanisms for clamping the work and cutting tools, of the lubricating and coolant systems, and of the electrical and hydraulic equipment; the dependability of all the protection devices is checked; and the effort required to operate the manual controls is measured. During these tests, the no-load power N_{ir} is to be measured at all speeds of the main drive (see Chapter 33 for N_{ir} measurements).

Idle-run tests are followed by load tests in which workpieces are machined at speeds and feeds enabling the maximum permissible loads to be attained, as well as momentary overloads of 25 per cent while employing up-to-date cutting tools of advanced design. In the load tests, the operation of all the mechanisms and systems of the machine tool is checked, as well as the operation of all clutches and brakes, and the dependability of the protection and safety devices. The power consumption N_c of the drive motor (see Chapter 33) is also measured. The results of these tests are assessed by the appearance of the machined surface (surface roughness, traces of vibration, etc.). The performance tests of machine tools, for which the piece output is indicated in the manufacturing specifications (single-purpose, production and automatic machine tools), should be conducted at the maximum output (in machining a typical workpiece). In many cases, metal waste is reduced by substituting tests using loading devices (see page 464) for performance tests based on metal cutting.

Next, the machine tool is checked against the accuracy and rigidity standards, and its vibration-proof properties are tested in regular machining operation (for details, see the subsequent chapters).

The general level of production techniques is checked and the quality of manufacture and assembly of lot-produced machine tools is more comprehensively assessed by supplementing the above-mentioned compulsory tests

with sample, or percentage, tests to which from 0.5 to 10 per cent of all the manufactured machine tools are subjected. Such sample tests may include, in addition to compulsory checks, measurement of the static rigidity of the principal units; more comprehensive tests of the vibration-proof properties, revealing the causes of the vibrations; measurement of the efficiency of the main drive; checking the temperature stability and variation in an idle-run test, etc.

The second type of machine tool tests are the pilot model tests that conclude the development of a new model of machine tool. These tests are conducted under laboratory conditions and the test results are given to an acceptance committee which decides whether they provide sufficient grounds for recommending lot production of the given model.

Recent investigations completed in ENIMS have enabled a detailed test procedure to be established for the pilot models of new machine tools. These tests have been systematized in accordance with the following principal criteria of machine tool appraisal:

1. An assessment of how well the machine tool complies with the stipulations of the design task, pertinent standards and the specifications, as well as the requirements of convenient and safe operation (including noise tests) and accessibility for repairs. As a rule, this includes an assessment of how correctly the degrees of versatility, mechanization and automation of the machine tool have been chosen.

2. An assessment of the capacity of the various mechanisms to ensure efficient, trouble-free operation in an idle run and under load, as well as of the service life of the machine in respect to wear and fatigue strength.

3. An assessment of the accuracy of operation of the machine tool.

4. An assessment of the production capacity of the machine tool. In the case of versatile or general-purpose machine tools, this assessment is made on the basis of the actually used effective power N_{ef}, taking into consideration the friction losses in the main drive and limitations resulting from insufficient vibration stability of the machine tool. In assessing the production capacity, it is necessary to place emphasis on the determination of the effectiveness of incorporating various devices which reduce the time required by handling operations that do not coincide in time with the formative process (for example: speed changing; clamping, loading and unloading the workpieces; measuring machined surfaces; etc.).

In order to properly assess pilot models, it is desirable that their quality indices be standardized, i.e., a definite quantitative evaluation be given for each index. However, a number of such indices still lack quantitative norms.

The procedure for conducting experimental research and tests in machine tool engineering is set forth in the subsequent chapters of Part Seven, and a number of questions concerning the operation of machine tools in regular service are also treated.

CHAPTER 24

TESTS AND INVESTIGATIONS CONDUCTED TO ASSESS MACHINE TOOL ACCURACY

The machining accuracy of a machine tool is characterized by the magnitude of the deviations in size, shape and relative positions of the elements of the surfaces obtained from the corresponding parameters of the given geometrical surfaces.

In recent years, the requirements made to the accuracy and surface finish of machined parts have considerably increased. For example, the permissible out-of-roundness of parts ground in precision cylindrical grinders is within 0.3 micron, while the surface irregularities should be within the values stipulated for a surface finish $\nabla 10$ through $\nabla 12$, according to USSR Std GOST 2789-59. The necessity for obtaining parts of a quality complying with such high requirements has brought investigations connected with the evaluation of machine tool accuracy to the fore. To check the accuracy of operation of a machine tool, it is necessary to know the character of the various factors that cause machining errors and the degree to which they affect the machining accuracy. At the present time, practically no accuracy standards of machine tool operation exist that give a single-valued definition of machining accuracy. This is due to the large number of such factors, the principal ones being: (1) geometrical (including kinematic) accuracy of the machine tool-fixture-cutting tool-workpiece complex, or system, taking into account the influence of clearances and the errors of the locating datum of the workpiece; (2) temperature deformations of the system; (3) processing rigidity which characterizes the deformation of the system under load; (4) stability of the system in setting up the work, traversing the units and during machining; (5) forced vibrations; and (6) dimensional wear of the cutting tool.

All of these factors, except the geometrical accuracy of the machine tool. are variable and, to some extent, controllable. Their influence on the accuracy of the workpiece can be reduced by changing the cutting speeds and/or feeds and other machining conditions, so that in the final analysis the attainable accuracy will be determined in general by the geometrical accuracy of the machine tool. The geometrical accuracy characterizes the quality of manufacture and assembly of the machine tool and, though it cannot qualitatively characterize the accuracy of a workpiece produced by the machine tool, it is one of the important characteristics of machine tool performance in this aspect.

The geometrical accuracy of each lot-produced machine tool should be checked. Norms of geometrical accuracy were first worked out by Dr.-Ing. G. Schlesinger who, in 1927, proposed a system of tests for determining the accuracy of manufacture of machine tools. This system has been used as the basis of the rules accepted in various countries for conducting machine tool accuracy tests, of the testing methods employed and, to a considerable extent, of the accuracy standards (permissible deviations) themselves. In the USSR machine tool accuracy is established by USSR standards (GOST). These so-called "Accuracy Standards" are based on the assumption that the geometrical errors of the given machine tool are constant systematic errors that are transferred fully to the workpiece. This assumption eliminates a very difficult procedure of making an analysis of the resultant error of the workpiece to determine the effect of geometrical errors of the machine tool, and replaces inspection of the workpiece by the corresponding geometrical accuracy tests of the machine tool. An analysis of the possible methods of shaping surfaces has enabled the interrelation to be cleared up between the errors of relative motion of the cutting tool and work in the machine tool on one hand, and the errors of geometrical features on the workpiece (errors in shape and relative positions of the surfaces) on the other hand. On this basis, a definite number of instrumental tests of geometrical accuracy have been worked out for each type of machine tool. These tests are usually conducted in the static state with the traverse and swivel movements of the machine tool units accomplished manually or at low speeds.

In conducting tests and research in the geometrical accuracy of machine tools, of prime importance are the conditions under which the measurements are made. The machine tool should be installed and levelled in the same way as for regular operation. Certain (not many) machine tools, mainly small ones with a very rigid bed, are installed for testing on three points of support. The great majority of machine tools are installed in the working position and levelled to spirit levels on a rigid foundation (testing facility) on more than three supports. The foundation bolts are not drawn up before the tests. Using wedges or adjustable shoes, the bed or base of the machine tool is positioned so that the deformation of its ways, measured by the spirit levels, is at a minimum. All travelling parts of the machine tool are put in their middle positions. The influence of the temperature on the results of the measurements is eliminated to the greatest possible degree. Accuracy tests usually follow the idle-run and load tests. The total volume of tests is determined by the corresponding accuracy standards stipulated in the pertinent USSR standards. Typical checks usually include accuracy tests on the geometrical shape of fitting (mounting) surfaces (straightness, flatness, out-of-roundness, taper, etc.); relative position of surfaces (parallelism, squareness, and alignment); the shape of the path of motion of operative members of the machine; co-ordinate positioning motions and kinematic chains.

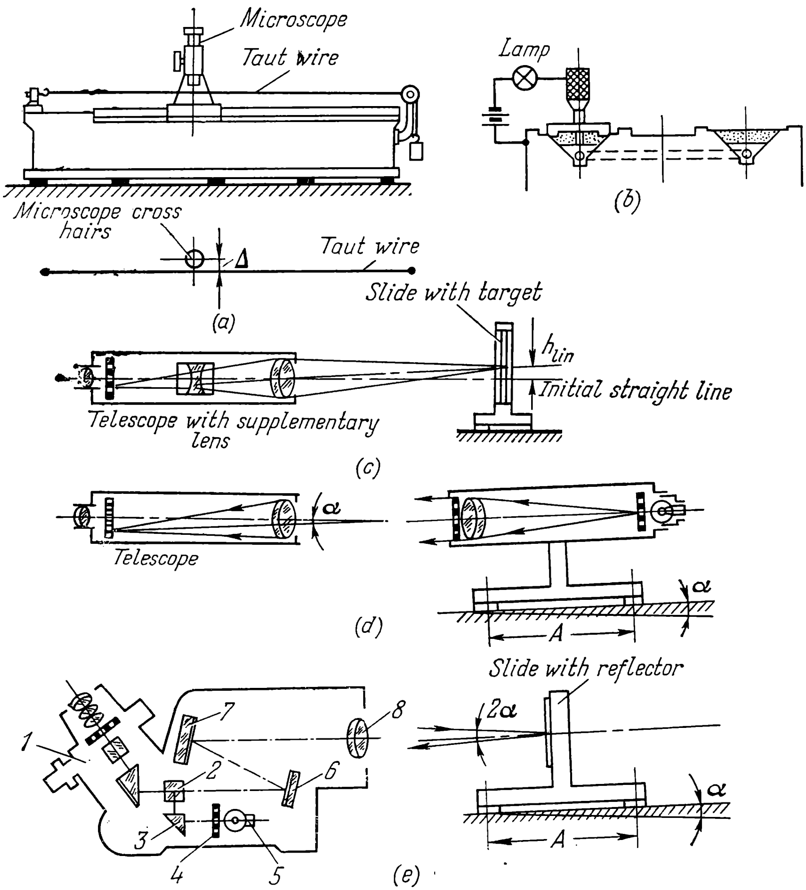


Fig. 284. Methods of checking the straightness of machine tool ways

The principal measuring tools used in geometrical accuracy tests are straight edges (wide-edge type from 500 to 3,000 mm long and toolmaker's straight edges up to 300 mm long); squares; gauge blocks; thickness gauges (the most widely used sets have blades ranging from 0.03 to 0.5 mm in thickness); stub or centre-type test mandrels (hardened to $52R_C$ and with a surface finish at least as good as ∇9); dial indicators with 0.01, 0.002 or 0.001 mm graduations; various types of levels with scale division values from 0.02 to 0.05 mm per 1,000 mm and, finally, optical instruments. The methods stipulated in the standards for conducting instrumental tests of the geometri-

cal accuracy are for arbitration purposes and, in certain cases, can be replaced by other methods ensuring no less accuracy of measurement. In Part Seven we shall consider only tests for straightness and for the accuracy of kinematic chains of machine tools in more detail since these tests are more complicated.

In accordance with the methodological principles worked out by ENIMS, the straightness of machine tool ways may be measured either by measuring linear values, determining the position of various sections of the ways in reference to an initial straight line, or by measuring the location of these sections in reference to one another, consecutively along the whole length of the ways.

In the first case, the straightness is determined by an optical sighting method (Fig. 284*c*); measurements along a taut wire (Fig. 284*a*); measurements made by means of a straight edge, gauge blocks and a thickness gauge; or by the hydrostatic method (Fig. 284*b*). The initial straight line, in reference to which measurements are made by these methods, is the optical axis of the alignment telescope, the horizontal projection of the taut wire, the working surface of the straight edge or the water level, respectively.

Depending upon the adopted method, the measuring instrument is a slide with the target (glass plate in the centre of which cross hairs are inscribed), microscope, set of gauge blocks or a hydrostatic head. This instrument is applied consecutively to the 0, 1st, 2nd, . . ., and n-th elementary plane surfaces along the ways. In each position, the deviation $h_{lin\ i}$ of the elementary plane surface on the way is recorded in reference to the initial straight line. In the general case, due to the misalignment between the direction of the way being tested and that of the initial straight line, the deviation $h_{lin\ n}$, measured on the last section, is the accumulated error. By calculating the constant error $b_i = \frac{h_{lin\ n}}{n} i$, referred to each section, where i is the number of the elementary plane surface, we obtain the deviation from straightness in the form $h = h_{lin} - b$. ENIMS recommends that the results of the measurements be set down in the following table:

No. of elementary surface i	Results of measurement $h_{lin\ i}$, microns	Constant error b_i, microns, referred to each section	Deviation from straightness $h_i = h_{lin\ i} - b_i$, microns

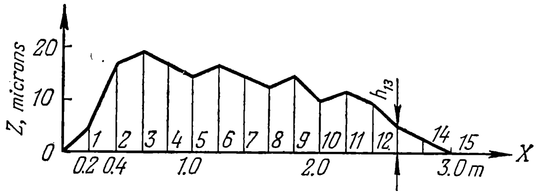


Fig. 285. Curve representing the actual shape of a way

It is advisable to plot a curve using the calculated values to obtain a scaled representation of the actual shape of the ways. An example of such a curve is illustrated in Fig. 285.

In the second case, straightness is checked with a level or by the collimation method.

Straightness testing by means of an alignment telescope and a collimator (Fig. 284*d*) will now be briefly treated.

A stand with a collimator, an illuminating device from which light emerges in parallel beams, is set consecutively on the 0 and 1st, 1st and 2nd, 2nd and 3rd, etc., elementary plane surfaces of the ways.

In the focal plane of the objective, the collimator has a graticule with a scale for reading the angle of inclination α of the axis of the collimator in reference to that of the telescope. In making the test the telescope is focused to infinity.

If one elementary surface is higher than the other, the collimator is tilted so that an angle α appears between the optical axes of the collimator and telescope (the optical axis of the latter is set parallel to the ways at the beginning of the test), and the cross hairs image of the collimator is displaced a distance $f\alpha$ in relation to that of the telescope, where f is the focal distance of the telescope objective and the angle α is given in radians. By moving the collimator stand along the ways, angle α is measured every 100 or 200 mm. The amount that one elementary surface is higher (or lower) than the next one is found from the formula $\Delta h_{ang} = A\alpha$, where A is the base length of the collimator stand. Then the distance from each elementary surface to the initial straight line (similar to h_{lin}) is

$$h_{ang} = \sum_{1}^{i} \Delta h_{ang}$$

Further procedure is as described above. If use is made of an autocollimator, combining the light source and eyepiece in a single housing, accuracy of measurement will be doubled (see Fig. 284*e*).

In an autocollimator, the image of the cross hairs on graticule *4*, placed in the focal plane of objective *8*, is directed through deflecting prism *3*,

beam splitter *2* and mirrors *6* and *7* to infinity in parallel beams from light source *5*. The beam of light falls on a mirror which is tilted, due to the lack of straightness of the way being tested, to an angle α in respect to the optical axis of the autocollimator. Thus, the image is reflected and is superimposed on the initial cross hairs with a certain displacement (in the eyepiece field of view) equal to $2f\alpha$, where f is the focal distance of the autocollimator objective. Micrometer eyepiece *1*, used in autocollimators, enables angle α to be read directly to 0.5 second of arc.

If the actual shape of one way has been determined, the shape of the other way is found from the data listed for the first way and from the results of measurement of the tilt angles of a bridge mounted on the two ways and moving along them. The tilt angles Θ are measured by a spirit level placed on the bridge. One way is higher than the other (for a given position of the bridge) by the amount $\Delta h = A\Theta$, where A is the distance between the ways.

The checking of the geometrical accuracy of a machine tool is very important, but it offers the user only an indirect guarantee of the machining accuracy since it does not take into consideration such a vital factor as the processing rigidity, on which machine tool accuracy under load depends. Therefore, with the aim of directly determining the machining accuracy, USSR standards provide for testing machine tool accuracy by checking the machined workpiece (performance tests) and, at the same time, checking the roughness of the machined surfaces. The type of workpiece, its material, kind of machining, speeds, feeds and depth of cut, clamping of the tool and other similar matters are assigned so that, in comparative tests, the influence of factors, not directly concerning the quality of machine tool manufacture, is reduced to a minimum. For example, the accuracy of the cutting tool and fixture is certified beforehand; the short duration of the tests excludes the influence of temperature deformations, etc. The deviation from the specified geometrical shape of workpieces machined at finishing speeds and feeds should be within the values stipulated in the corresponding USSR standards. The machined surface should be clean and without traces of chatter and vibration.

CHAPTER 25

CHECKING THE ACCURACY OF KINEMATIC CHAINS OF MACHINE TOOLS

In cases when the surface of the workpiece in a machine tool is produced by one or several complex formative motions, the machine tool should have, in addition to an adequate geometrical accuracy, a high kinematic accuracy. Kinematic accuracy, in the sense here referred to, means the accuracy with which the velocity relationships are maintained for the operative members of a machine tool that participate in producing some complex formative motion. Kinematic errors of a machine tool result from the errors in the internal kinematic chains. In the manufacture of a new model of machine tool, as well as in operation and after repairs, it is necessary to know the kinematic accuracy. To this end, the standards make provisions for testing the corresponding internal kinematic chains of gear-cutting machines, engine lathes, relieving lathes, thread-milling machines and thread grinders. These tests are conducted together with the accuracy tests.

The accuracy of the thread-cutting trains of engine lathes and similar machines is usually tested with the machine running. Here the contact point of the measuring instrument touches a thread of a master screw mounted between the centres of the machine being tested, which has been set up to the pitch of this screw. The measuring instrument, which may be equipped with a recording device, is mounted on the carriage of the machine tool (lathe).

In more detail we shall consider the methods of checking the kinematic accuracy of gear-cutting machines, for which the kinematic accuracy is one of the most important features.

The kinematic errors $\Delta\varphi$ of the machine are shown up on the workpiece as pitch and tooth profile errors and the accumulated errors of the base and circular pitches. To assess the capacity of the machine to produce high-quality gears, it is necessary to check its kinematic accuracy. This enables the causes of the kinematic errors to be revealed and eliminated.

The simplest method of checking the kinematic accuracy is with the use of a theodolite. For this purpose, the machine is set up to cut a gear with the maximum possible number of teeth (within the capacity of the machine). The theodolite is set in the centre of the machine table and is sighted on a collimator tube stationarily mounted adjacent to the machine. Then the spindle of the machine is turned one revolution. At this, the table with the

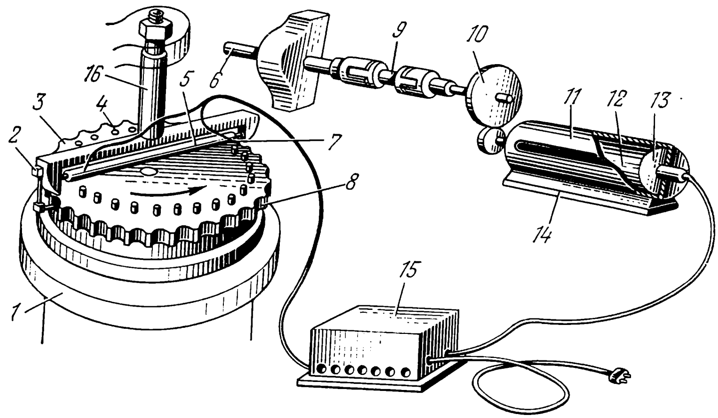


Fig. 286. Principle of the TSNIITMASH kinematometer

theodolite turns through the angle φ which is theoretically equal to

$$\varphi_t = 2\pi i \text{ radians}$$

where i is the gearing ratio of the kinematic train being tested. Then the telescope is turned back to its initial position in which its optical axis coincides with that of the collimator, and the actual angle φ, through which the table has turned, is read off on the horizontal circle scale of the theodolite. The difference $\Delta\varphi = \varphi_t - \varphi$ is the kinematic error in the angle φ of table rotation. This is repeated until the table has rotated through 360°. The accuracy of this method may reach 1 or 2 seconds of arc, but the periodic stopping of the machine, impermissible in cutting precision gears, does not comply with the conditions under which errors should be revealed in regular operation of the machine.

There is no such drawback in the method developed by TSNIITMASH (Central Research Institute of the Heavy Engineering Industries of the USSR) using a kinematometer. The test is conducted in an idle run. This instrument (Fig. 286) consists of three parts: transmitter *16*, electrical unit *15* and indicator *14*. The transmitter, mounted on and rotating together with machine table *1*, is a disk *3* along whose circumference equally spaced and parallel pins *4* are arranged. Suspended above the transmitter on fixed centres is bar *5* at whose ends blades *7*, insulated from each other, are provided. As the table rotates, the blades, contacting diametrally opposed pins, close the electric circuit and periodically light inertialess lamp *13* located inside the indicator. After contact is made, the bar with the blades is automatically deflected by a pusher driven by a special device *2* which is actuat-

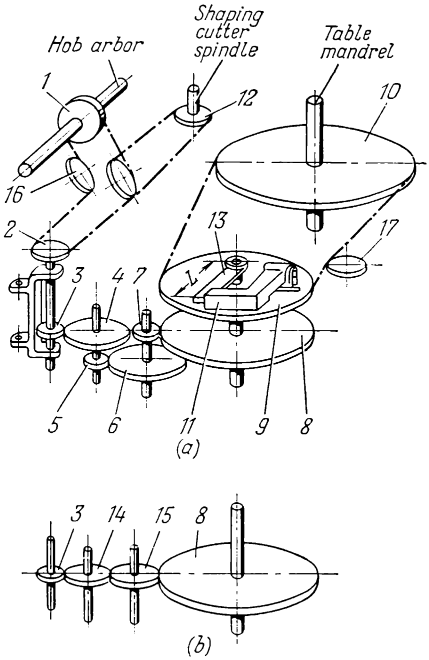


Fig. 287. Principle of the Levashov synchronometer:
1, *2*, *9*, *10* and *12*—input disks; *3*, *4*, *5*, *6*, *7*, *8*, *14* and *15*—friction drive disks; *11*—recorder; *13*—measuring lever; *16* and *17*—guide rollers

ed by cam disk *8*. At this, the pin moving in the direction of blade deflection passes under the blade.

The indicator consists of stationary cylinder *11*, having a straight through slit with a scale, and internal rotary cylinder *12* having a narrow helical slit in the cylinder wall. Cylinder *12* is linked to hob arbor *6* of the machine through cardan shaft *9* and gearing *10* which increases the angular velocity of cylinder *12* a whole number of times in relation to that of the hob arbor.

As the transmitter sends a signal, inertialess gas-discharge lamp *13* produces a flash, lasting some thousandths of a second, which the observer sees at the point of intersection of the slits in the two cylinders. Noting on the scale the positions of consecutive spots of light, it is easy to determine their displacement from each other. This displacement is proportional to errors in the angular displacements of the table. Possessing a sensitivity up to 1 second of arc and lacking the shortcomings of the first method, this instrument, however, measures only discrete values of the function of the relative motion of the operative members of gear-cutting machines. To analyse the errors of the kinematic chain, these values must undergo comparatively complex mathematical treatment. This is a significant disadvantage of this method. Moreover, the behaviour of the machine within the intervals limited by the transmission of the discrete signals remains uninvestigated.

At the present time, a continuous testing method, developed by A. Levashov at the Komsomolets Plant, has found application in practice. The operation of the synchronometer designed by Levashov (Fig. 287*a* and *b*) is based on kinematically closing the chain being tested by means of another, parallel chain made in the form of disks linked together by friction or by tapes. The synchronometer can be used to check any trains with gearing ratios ranging from 1 : 1 to 1 : 250. It takes from 30 sec to 4 min to make one diagram (for one table revolution), depending upon the setup of the machine. Using the more sensitive electric recorder, type БВ-662, with inductive

pickup, type БВ-785, instead of a lever-type recording device, it is possible to obtain a magnification of 5,000X. The results of the test are in the form of a continuous diagram which gives a precise record of the errors in the elements making up the kinematic train. By harmonic analysis of curves obtained on the diagram during one revolution of the table, it is possible to locate the source of the error.

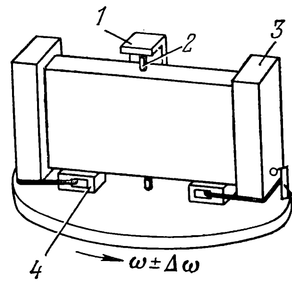


Fig. 288. Instrument for measuring the kinematic accuracy of machine tools by the seismic method

Among efficient methods for continuously testing the kinematic accuracy is the seismic method developed in the Higher Technical School of Aachen. An instrument (Fig. 288) based on this principle consists of a housing carrying supports *1* to which a very flexible torsion spring *2* is attached. Suspended from the spring is heavy mass *3*. The instrument is mounted on the machine table. Upon uniform rotation of the table, the relative position of the housing and the suspended mass remains unchanged. Kinematic errors of the train being tested upset uniform table rotation which leads to rotation of the suspended mass, due to inertia, in respect to the housing. This rotation is sensed by some type of contactless measuring pickup *5* (for example, capacity pickup or variable inductor) and is transmitted to the recording device. Damping magnets *4* are used to damp the natural oscillations of the measuring system. Since the natural frequency of vibration of mass *3* on the spring is very low, the mass moves practically uniformly. An amplifier and an oscillograph enable the pickup readings to be magnified up to 30,000 times. This corresponds to a measuring accuracy of 0.1 micron on a diameter of 4,000 mm.

A drawback of the seismic method is the impossibility of measuring frequencies lower than 0.2 or 0.3 cps, i.e., frequencies lower than the natural frequency of vibration of seismic mass *3*. The instrument can measure all deviations in angular velocity of the elements making up the kinematic train, except those caused by the accumulated pitch error of the index worm wheel (this error appears once in each revolution of the machine table). This lowers the value of this method.

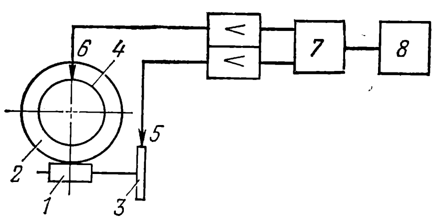


Fig. 289. Schematic diagram of the measurement of the ratio of index worm gearing by the "magnetic scales" method

The "magnetic scales" method, proposed by the Prague Research Institute of Machine Tool Engineering for checking the uniformity of rotation of the index worm gearing in gear-cutting machines, is based on a principle resem-

bling linear measurement with the aid of the wave length of light. The circular magnetic scale is a metal disk coated with a layer suitable for making magnetic recordings. Waves of a definite frequency are recorded on the disk. A special method of recording enables a high uniformity of wave length to be obtained (the maximum accumulated error is within 1 second of arc).

Let us consider the arrangement for measuring the gearing ratio of index worm gearing (Fig. 289). Circular scales *3* and *4* are mounted on the shafts of worm *1* and worm wheel *2*. The number of magnetic waves on the disks corresponds to the ratio of the gearing being tested. Upon operation of the gearing, the recording is read off by magnetic heads *5* and *6* whose signals are transmitted through an amplifier to phase meter *7* and then to the recording device (oscillograph) *8*. Phase shifts in the two signals being read, sensed by phase meter *7*, determine the amount of nonuniformity of rotation of the elements in the gearing. The method enables a relative error as small as 10^{-7} to be rapidly measured. It can also be employed for checking the kinematic error of the generating and indexing trains of gear-cutting machines. However, with all its advantages, the method cannot be used to record errors caused by high-speed members of the train (frequency over $\simeq 4$ cps). This is due, to some degree, to the minimum wave length ($\simeq 20$ microns) that can be satisfactorily recorded on a circular magnetic scale.

CHAPTER 26

TEMPERATURE AND THERMAL DEFORMATION INVESTIGATIONS IN MACHINE TOOLS

It has been previously mentioned that the generation of heat and the resulting thermal deformation are factors leading to machining errors. In respect to a large group of parts and units, such as bearings, ways and others, their heating is one of the criteria of their performance. Heating may be due to external sources of heat (sun rays, heating facilities, etc.) and sources within the machine tool. At a constant ambient temperature, the temperature of the machine tool is raised by heat generated in the cutting zone and heat evolved in the operation of the mechanisms and systems of the machine. The relative significance of these sources of heat generation may vary and depends to a great extent upon the type and construction of the machine tool. The cutting process has only a slight influence on the thermal deformation of machine tools for finishing operations due to the small machining allowances.

Thermal deformation of a machine tool is made up of the deformation of its component parts and frequently continues to grow until it reaches substantial values. According to data of ENIMS, for example, after a continuous idle run for 5 hours at a speed of approximately 1,200 rpm, the spindle nose of the model 1A62 lathe was displaced almost 20 microns in a horizontal plane and 80 microns in a vertical plane.

Being a systematic variable error, thermal deformation affects the accuracy of linear dimensions of the workpiece, of its geometrical shape and of the relative positions of its surfaces. While the first of these can be easily compensated for by the operator in general-purpose machine tools (the heating process in these machines takes from 4 to 8 hours) and by additional adjustments in automatic machine tools, it often proves impossible to compensate for the other two kinds of errors. As a result, the workpiece obtained may not be within the limits of the tolerances specified for the size and surface finish. Consequently, in designing precision machine tools it is necessary to be able to determine the magnitude and direction of the thermal deformation of the various component parts and units, the time required for temperature stabilization in heating, and the configuration of the temperature field (the whole complex of instantaneous temperature values at various points) to reveal the heat sources (places heated to the greatest degree).

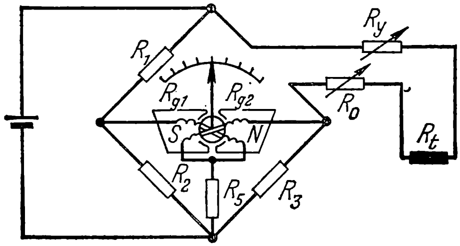


Fig. 290. Unbalanced single bridge circuit with a logometer for temperature measurement with a resistance thermometer

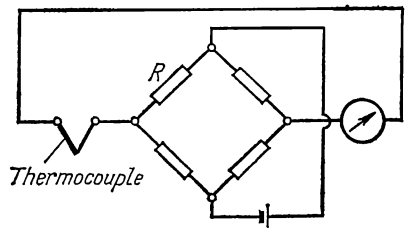


Fig. 291. Thermocouple circuit for temperature measurement with compensation for the heating of the free ends of the thermocouple

The main instruments used for temperature measurements in machine tools are resistance thermometers and thermocouples.

The action of a resistance thermometer is based on the property of metals and semiconductors to change their electric resistance upon a change in temperature. In many cases, the thermistor (the sensitive element of the thermometer) is a thin wire from 0.02 to 0.06 mm in diameter and 5 to 50 mm long, wound on a body of insulating material and protected by a hood. The wire material should possess the largest possible and constant temperature coefficient α and, at the same time, the highest possible resistivity. Most widely applied are thermistors of platinum (t up to 500°C), copper (t up to 180°C) and nickel (t up to 250° or 300°C). Semiconductor thermistors, of the MMT, TOC and other types, are also used. Their temperature coefficient α is from 8 to 10 times larger than that of metals. Of small size (tenths of a millimetre) and high stability, sensitive elements of semiconductors are very convenient for measuring "points" on a surface. This is especially important when there are large temperature drops. To measure the temperature. the thermometer is connected to a suitable measuring circuit. Most extensively employed are circuits with unbalanced single bridges having a magnetoelectric logometer for the measuring facility (Fig. 290). Three arms of the bridge are manganin resistors R_1, R_2 and R_3. The fourth arm is made up of the thermistor R_t and the resistors R_0 and R_y which serve to balance the bridge prior to measuring. The logometer consists of two frames with coils R_{g1} and R_{g2} fastened rigidly together and secured on a common shaft in the field of a permanent magnet. If the currents passing through coils R_{g1} and R_{g2} are the same, the logometer hand indicates zero. The value of R_t is changed in heating, thereby unbalancing the bridge and leading to a reduction of the current in one of the frames and rotation of the frames with the hand. A part of resistor R_5 is made of copper, enabling the temperature

error of the logometer to be compensated for. Within known limits, the logometer reading does not depend upon the voltage of the power supply.

A more universal and convenient means of measuring the temperature in machine tool research is a thermoelectric thermocouple transducer, consisting of two dissimilar conductors, soldered together at one end (thermal junction) and connected to the measuring instrument at the other. Upon heating the thermal junction of the thermocouple, current will flow in the measuring circuit with a thermoelectromotive force that depends only upon the material of the thermocouple and the difference in temperature of the thermal junction and the free ends of the conductors. The most widely used thermocouples—copper-constantan, iron-constantan and nichrome-constantan—provide a thermoelectromotive force of several millivolts at 100°C. Therefore, in measuring the temperature with a thermocouple, there must be a very sensitive millivoltmeter in the circuit. Various methods are used to compensate for the heating of the free ends of the thermocouple. One is shown in Fig. 291. Here the thermocouple and millivoltmeter are connected into the measuring diagonal of the bridge whose arms, except for copper arm R, are made of manganin wire. The resistor R is placed near the free ends of the thermocouple. In normal operation of the thermocouple, the bridge has no effect on the reading of the millivoltmeter. Upon an increase in the temperature of the free ends of the thermocouple, the value of R also increases, the bridge is unbalanced and the voltage appearing across the measuring diagonal compensates for the reduction in thermoelectromotive force of the thermocouple. The accuracy of compensation may reach 0.04 mV per 10 deg C.

The method of mounting and fastening the thermocouple on the parts of the machine tool may substantially affect the measuring accuracy. Some of the fastening methods are illustrated in Fig. 292.

Thermal deformation is measured by micron dial indicators, mikrokators and spirit levels with scale graduations not coarser than 0.02 mm per 1,000 mm. Uprights and bridges on which the measuring instruments are mounted should be carefully heat-insulated to avoid distortion of the results of measurement. Frequently, thermal deformation is measured by the effect it has on the workpiece. This is done by machining (for example, in a cylindrical grinder) specially prepared blanks after definite intervals of time and then measuring their dimensions.

The units of the machine tool are not moved mechanically in the course of the experiment and infeed is accomplished only due to thermal displacements. The experiment continues until a steady temperature is reached.

If other factors affect the results of the experiment, also causing systematic variable errors, for example, dimensional wear of the cutting tool, then the experiment is conducted as follows. Several lots of blanks, carefully selected for quality and having strictly maintained machining allowances,

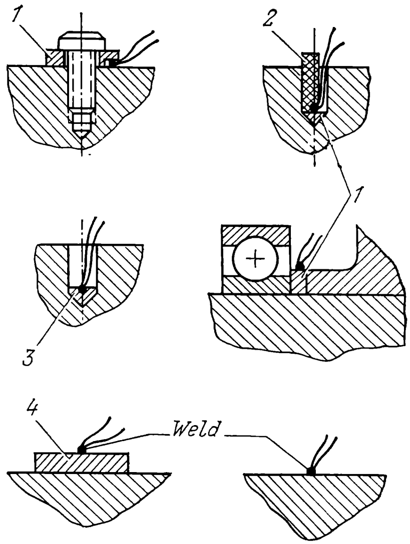


Fig. 292. Methods of securing the thermocouple for measuring the temperature of machine tool components:

1—lead washer; 2—plug of laminate fabric base; 3—fusible metal; 4—plate with high heat conductivity

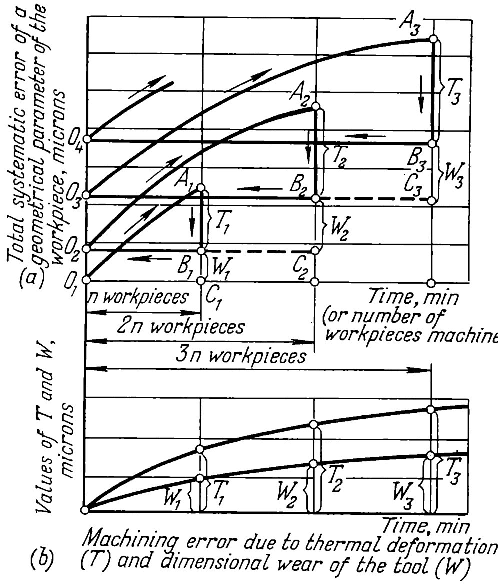


Fig. 293. Variation of the systematic variable machining errors with time

are machined. Each lot consists of *in* blanks, where *i* is the number of the lot in the order they are to be machined. An accuracy diagram (Fig. 293*a*) is plotted from the results of machining each consecutive lot, and the experiment is discontinued until the machine tool cools down to the ambient temperature, after which the next lot can be machined. Plotted along the axis of ordinates on the accuracy diagram is the total systematic error of some geometrical parameter of the workpiece (for example, the diameter) caused principally by the thermal deformation and the dimensional wear of the tool. Curve O_1A_1 indicates the gradual increase in the total systematic error in the process of machining the first lot. At point A_1 the test was discontinued and the machine tool was permitted to cool down to the initial temperature. This made it possible, in machining the first blanks of the second lot, to single out the influence of the dimensional wear of the tool (intercept B_1C_1) on the total error. The remaining part—intercept A_1B_1—represents the share of the thermal deformations in the total maximum error of the workpieces of the first lot. Proceeding in a similar manner in machining the

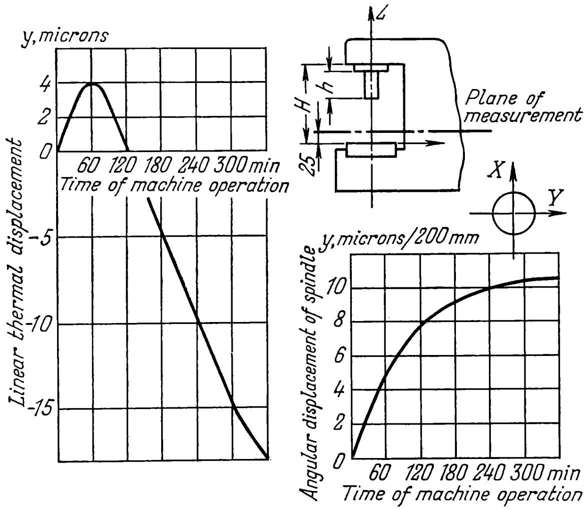


Fig. 294. Curves showing the linear and angular thermal displacements of the spindle axis in the model 2B440 jig borer (width of the working surface of the table—400 mm; $H = 350$ mm; $h = 110$ mm and $n_{sp} = 1{,}000$ rpm)

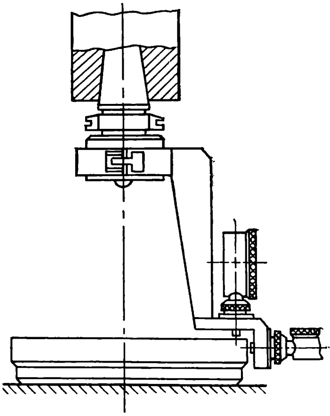


Fig. 295. Instrument for measuring linear and angular displacements of the spindle of a jig borer that are due to thermal deformation

other lots of blanks, it is possible to plot curves showing the variations with time of the machining errors caused by the thermal deformation and wear (Fig. 293*b*). If the influence of thermal deformation is considerable, this deformation should be investigated in detail. For this purpose, simultaneous measurements are made of the temperature field and of the magnitude and direction of the thermal deformation of the principal units and parts. The results of these measurements are plotted in curves similar to the one shown in Fig. 293*b*. To determine the influence of the various sources of heat on the magnitude of temperature deformation, the same experiments are conducted but with consecutive heat insulation of these sources (for example, with asbestos packing).

The investigation of temperature and thermal deformation in machine tools has enabled measures to be worked out for reducing the effect of heating on the machining accuracy. Such measures include the manufacture of machine tool components of materials with a low coefficient of linear expansion (for example, invar); compensation of thermal deformation by special mechanical devices; stabilization of the shop temperature; and measures for reducing the amount of heat generated by the machine tool itself (effective cooling of the electric motor; arranging the hydraulic pumping station and

tank outside of the machine tool; raising the accuracy of the spindle bearings, providing minimum interference in them and using mist lubrication; reducing the number of gear transmissions in the speed gearbox, etc.).

A more comprehensive evaluation of the accuracy of a machine tool is obtained if thermal deformation is taken into account in conducting the accuracy tests. Norms of temperature stability have been worked out in ENIMS* for jig borers and cylindrical grinders. They supplement the accuracy standards of these machines. These norms are based on the generalization of the results obtained in the measurement of thermal displacements in Soviet machine tools. Given in Fig. 294 are the results of such measurements obtained in testing the lot-produced jig borer, model 2B440. The machine ran idle at a spindle speed of 1,000 rpm. The temperature of the air was maintained constant in the room where it was installed. Measurements were made every 15 or 20 min in the course of 6 hours. This provided data for plotting curves of the linear and angular displacements of the spindle axis in accordance with the time the machine had been in operation. For this purpose, a test disk with precise cylindrical and end surfaces was clamped on the machine table square to the spindle axis. A quick-change holder with two micron-type dial indicators was mounted on an arbor clamped in the spindle (Fig. 295). In turning the spindle by hand, the first indicator measured the lack of squareness of the spindle axis to the flat surface of the disk (or the table) at diameter D, while the second indicator measured the misalignment between the spindle and the table. After measuring the deviations at two diametrically opposed points before heating (y_{1cold} and y_{2cold}) and after heating (y_{1hot} and y_{2hot}), the standardized linear Δ_{tl} and angular Δ_{ta} displacements of the spindle axis and table in plane YOZ were determined using the formulas

$$\Delta_{tl} = \frac{(y_{1hot} - y_{1cold}) - (y_{2hot} - y_{2cold})}{2} \text{ microns}$$

$$\Delta_{ta} = \frac{(y_{1hot} - y_{1cold}) - (y_{2hot} - y_{2cold})}{1{,}000\,D}$$

where y_{1hot}, y_{1cold}, y_{2hot} and y_{2cold} are given in microns and D in mm. The same deviations in plane XOZ turned out to be considerably smaller.

* Under the supervision of V. Alferov and Yu. Sokolov.

CHAPTER 27

STATIC RIGIDITY INVESTIGATIONS IN MACHINE TOOLS

When a blank is being machined in a machine tool the cutting force does not remain constant. This is due to the variations in the cross section of the undeformed chip, machining allowance and the mechanical properties of the material. The cutting force is also changed by dulling and wear of the cutting tool, formation of a built-up edge and certain other factors. The action of the varying cutting and friction forces, in conjunction with the corresponding action from the drive motor, deforms the elastic elements of the machine tool, thereby changing the cutting and friction conditions or the operating conditions of the drive motor. Thus the circuit of action is closed (see Part Five, Chap. 12), and the actions, called linkages, are directional. The circuit determining the behaviour of the whole system of the machine tool under the given conditions is called the main linkage circuit. Most often the main linkage circuit of the dynamic system of a machine tool in cutting is an "elastic system—cutting" circuit since the flat characteristic of an induction motor of ample power rating and the very slight change in the friction in the antifriction spindle bearings (except in jamming of the rolling elements) upon a variation in the deformation enable all other linkage circuits to be neglected.

In order to account for the great variety of phenomena occurring during machine tool operation, including errors in the size and shape of the workpiece, it is necessary to know the properties of the elements making up the closed system of the machine tool. These properties can be characterized by the relationship between the effect of the preceding element on the given element (input co-ordinate) and the result of this effect (output co-ordinate), i.e., the characteristic of the element. If the input co-ordinate is constant in time, the characteristic will be static; if it varies in time, the characteristic will be dynamic.

In a special case (of great practical interest, however), the static characteristic of an elastic system, which includes all the parts and units of the machine tool, the workpiece and the cutting tool, is the ratio of the displacement y of the point (nose) of the cutting tool in a direction normal to the machined surface to a force P which imitates the constant cutting force. This

characteristic

$$k = \frac{y}{P} \text{ mm per kgf (or microns per kgf)}$$

is called the *unit deflection* (or compliance).

The reciprocal of k, called the *rigidity*, is frequently used in practice. Thus

$$j = \frac{P}{y} \text{ kgf per mm (or kgf per micron)}$$

The rigidity is one of the primary criteria of machine tool performance since it determines the accuracy under load in steady state operation (static error). The less the rigidity of a system, the larger the shape and size errors of the machined workpiece, i.e., the lower the machining accuracy.

The first fundamental investigations in rigidity were conducted in the USSR by K. Votinov as far back as 1935. Problems of machine tool rigidity were further developed in the works of Soviet scientists A. Sokolovsky, D. Reshetov, B. Balakshin, Kh. Enikeyev and others. Investigations showed that the total deformation y of a system subject to the action of force P depends to a greater degree upon the deformation in the joints than in the parts of the machine tool themselves. Deformation in joints depends to a great extent on the surface quality (accuracy of shape and roughness), i.e., on the processing techniques. Therefore, to assess the quality of the manufacture of a machine tool as well as its design, rigidity tests should be carried out in addition to the geometrical accuracy tests. Machine tools are complex systems. Depending upon variations in the magnitude and direction of the cutting forces and the positions of the travelling units, various joint surfaces may carry the loads. Consequently, there will be different values of the rigidity.

Therefore, in order to obtain sufficiently reliable and objective rigidity test results, it is necessary to conduct tests that approximate as near as possible the more typical real cases of machining, retaining, however, static loading to simplify the tests. For this purpose, a whole complex of special problems must be solved first.

1. The direction of the loading force is chosen on the basis of an analysis of the magnitudes of angles α and β, where $\alpha = \tan^{-1}\frac{P_y}{P_z}$ and $\beta = \tan^{-1}\frac{P_x}{P_z}$ (here and further on, P_x, P_y and P_z are the axial, horizontal and vertical components of the cutting force, respectively). Depending upon the material being machined, the tool geometry and other factors, angles α and β may vary in fairly wide ranges. However, as has been established by investigations conducted in ENIMS, at $\alpha = 13°$ to $22°$ or less, the horizontal displacement of the tool in a lathe is equal to zero. Therefore α should not be less than

25° or 30°. To simplify the tests it has been assumed that $\beta = 0$, since the rigidity varies very little with a variation of β within the range of most typical values. In connection with the negligible effect of the torque on the machining accuracy, its influence is not taken into account. Finally, all this enables a lathe to be loaded by the resultant of forces P_y and P_z which passes through the line of centres (since the workpiece diameter only slightly affects the rigidity) at an angle of $\alpha = 30°$. Such simplification of the loading scheme has no practical effect on the test results. A similar procedure is employed for other types of machine tools.

2. The magnitude of the loading force should be sufficient to cause a displacement that can be precisely measured with an ordinary dial indicator reading to 0.01 mm, but it should not exceed the load permissible for the machine tool being tested. In the work of V. Vedensky, the value of the loading force is expressed as a function of the principal dimension of the machine tool. Thus, for lathes, $P = 0.75\, D^{1.5}$ kgf, where D is the maximum diameter (mm) of workpiece accommodated in the lathe.

3. The co-ordinates of the point of application of the force are chosen in accordance with a typical case of machining and on the basis of convenient arrangement of the instruments. Taking into consideration the fact that the position of the cross slide and a variation in tool overhang from h to $2h$ (h is the height of the tool) have only a slight influence on the rigidity, ENIMS recommends the distance from the point of force application to the toolholder (or square turret) to be taken as $H = \sqrt[3]{D^2}$ mm for lathes. In tests with the work held in a chuck, the position of the point of force application at the headstock is determined by the overhang and diameter of the centre. The overhang of the tailstock spindle is stipulated for tests with the work held between centres (Fig. 296).

4. A definite typical arrangement of the travelling units on the machine tool is recommended (Fig. 296).

5. The machine is installed and levelled as for regular operation. Prior to the test, units that are stationary during operation are secured; the adjustment of the movable associations is checked as well as the tightness of the fixed joints.

6. The quality of adjustment of the joints can also be assessed if the requirement is complied with that the load is applied immediately after setting the units of the machine in the specified positions.

On the basis of the requirements as to the direction and magnitude of the loading force and the co-ordinates of its point of application, ENIMS has developed a number of universal instruments for measuring the rigidity of various machine tools. The principal components of each such device are the loading facility (jack and a previously calibrated dynamometer with a load indicator) and displacement indicators. To reduce errors in the direction of the loading force and in the co-ordinates of its point of application,

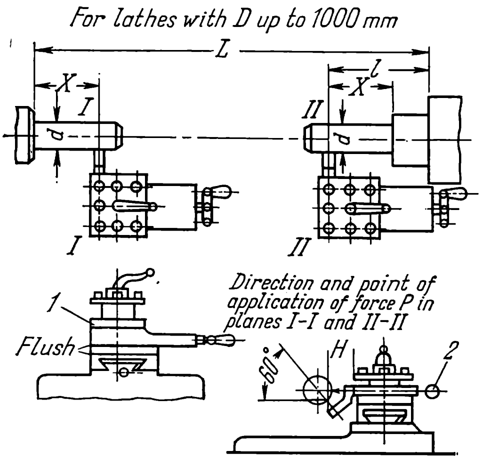


Fig. 296. Rigidity test conditions for general-purpose lathes in accordance with USSR Std GOST 7895-56:

1—loading device; 2—dial indicator for measuring relative displacements

For Lathes with a Maximum Diameter of Workpiece $D = 400$ mm

H	l	d	L
50	140	40	0.75 to 0.85 of the maximum distance between centres

Taper hole in headstock or tailstock spindle	X
No. 4 Morse taper	55
No. 5 Morse taper	70
No. 6 Morse taper	90

With an applied force of $P = 560$ kgf, the maximum permissible displacement, mm, of the toolholder (square turret) in reference

to the mandrel in the headstock spindle 0.21

to the mandrel in the tailstock spindle 0.27

the force is applied to the elements of the machine tool through an intermediate member with a spherical surface. Shown as an example in Fig. 297 is the instrument developed by ENIMS for measuring the rigidity of lathes.

Integrated indices of rigidity are employed in production tests of machine tools. They determine the overall rigidity of the machine, characterized by the displacement of the point of the cutting tool in reference to a rigid workpiece when the cutting forces are applied. The overall rigidity can be found as the arithmetic mean of the rigidity values obtained for several (four or five) load stages. The magnitudes of the stages depend upon the chosen maximum force (see page 407). The test is repeated 2 or 3 times to ensure more reliable results of the measurements.

As norms of rigidity with which the measured values of overall rigidity can be compared, the USSR standards list indices obtained as a result of the generalization of statistic data on the actual rigidity of produced machine tools. These norms represent rigidity values that are quite attainable and, at the same time, enable the quality of the machines being tested to be maintained at a high level. V. Vedensky (ENIMS) expressed the rigidity in the

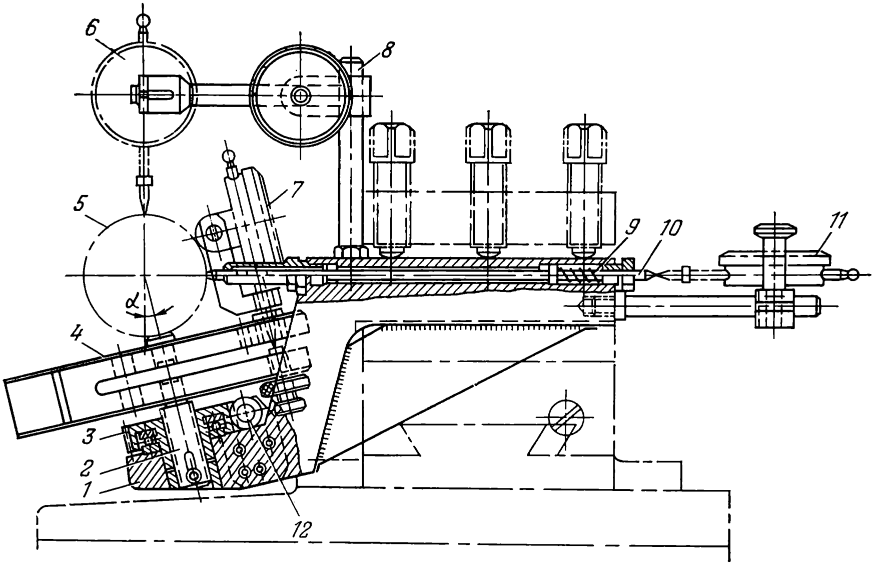


Fig. 297. Instrument for measuring the rigidity of lathes under shop conditions:
1—body; *2*—screw; *3*—nut; *4*—dynamometer; *5*—mandrel; *6*—dial indicator for measuring displacement along the z-axis; *7*—dial indicator of the dynamometer; *8*—vertical post; *9*—spring; *10*—pin; *11*—dial indicator for measuring displacement along the y-axis; *12*—worm

form of a power function of the principal dimension of the machine tool. For example, this empirical relationship has the following form for lathes:

$$j = 180\sqrt[3]{D} \text{ kgf per mm at the headstock}$$

and

$$j = 140\sqrt[3]{D} \text{ kgf per mm at the tailstock}$$

where D is the maximum diameter of workpiece accommodated in the lathe.

To simplify checking under shop conditions, the rigidity norms indicate the maximum permissible displacement of definite parts of the machine tool when the maximum loading force is applied (see Fig. 296).

The static method of determining the overall rigidity of machine tools, used in many Soviet plants, is simple but, as any other static method, it does not take into consideration all the specific features of the cutting process. A sufficiently simple and efficient method of determining machine tool rigidity in the cutting process has not yet been developed, though the search continues.

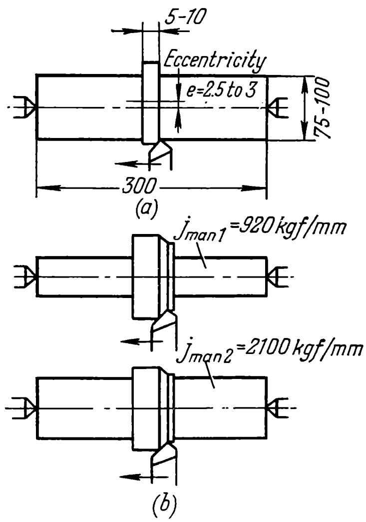


Fig. 298. Kinds of blanks used to determine lathe rigidity in cutting

Existing methods are based on the machining of an eccentric or stepped blank in a single pass. Consequently, the depth of cut sharply changes in the course of machining, leading to an instantaneous increase in the cutting forces and a displacement of the tool together with the carriage. Errors in the geometrical shape of the workpiece appear, as a result, and the rigidity of the machine tool is assessed on the basis of their magnitude. One such method was developed in the Kalinin Polytechnical Institute in Leningrad under the supervision of A. Sokolovsky. Here, a blank with an eccentric shoulder is turned in a lathe (Fig. 298*a*) with a minimum depth of cut of 0.1 or 0.2 mm. The dimensions of the blank and the tool geometry are taken the same for various tests (or experiments) so that comparable results will be obtained. By determining the runout Δ_{bl} of the blank before it is turned (of the order of 5 or 6 mm) and the runout Δ_{wk} of the turned workpiece, the so-called "accuracy improvement" $\varepsilon = \frac{\Delta_{bl}}{\Delta_{wk}}$ is found and the rigidity is calculated by the empirical formula

$$j = \lambda C_P s^{0.75} \varepsilon \text{ kgf per mm}$$

where $\lambda = \frac{P_y}{P_z}$

C_P = coefficient from the formula for determining the vertical component of the cutting force (given in cutting speed and feed handbooks)

s = rate of feed, mm per revolution.

Depending upon the tool geometry, condition of the cutting edge and the workpiece material, values C_P and λ may vary in fairly wide ranges. Thus, for the usual plan approach angles φ, the value of λ varies from 0.3 to 0.65, while for engineering steels $C_P = 144$ to 205 and for cast iron $C_P = 100$ to 150. The necessity for precisely determining the values of λ and C_P, to comply with the actual production conditions, makes this method either too cumbersome or extremely inaccurate (error up to 50 per cent, or even more), whereby it cannot be recommended for use. The search for new shop methods of determining the rigidity continues. Thus, a method proposed in the Zaporozhye Mechanical Engineering Institute involves the use of two man-

drels. In this method, the results of the experiment, characterizing the rigidity of the machine tool, do not directly depend upon the grade of the material being machined, cutting speeds and feeds or the tool geometry, which are maintained constant in various experiments. Two stepped, entirely identical specimens, mounted on two mandrels of different rigidity (Fig. 298*b*), are machined consecutively in the machine tool. The machining conditions (tailstock spindle overhang, tightness of the tailstock clamping nuts, tool setting, etc.) are identical for the two specimens. Assuming that the cutting forces in machining on the two mandrels are equal, we can write

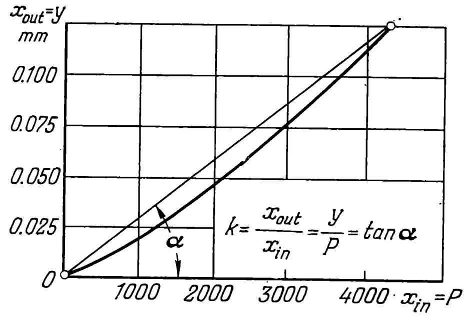


Fig. 299. Experimental static characteristic of the elastic system of the model 1Д62M engine lathe

$$\frac{y_1}{\frac{1}{j}+\frac{1}{j_{man1}}}=\frac{y_2}{\frac{1}{j}+\frac{1}{j_{man2}}}$$

where y_1 and y_2 = differences in radii on the stepped specimens on the two mandrels

j = rigidity of the machine tool

$j_{man\,1}$ and $j_{man\,2}$ = rigidities of the two mandrels (according to data from the Institute, 920 and 2,100 kgf per mm, respectively).

The rigidity of the machine tool then is

$$j=\frac{(y_1-y_2)\,j_{man1}\,j_{man2}}{j_{man2}\,y_2-j_{man1}\,y_1}\text{ kgf per mm}$$

Numerous other methods have been devised for assessing the overall rigidity of machine tools.

In conducting rigidity tests under laboratory conditions, it is possible to plot a curve of the deflection as a function of the load, i.e., $y=f(P)$ as in Fig. 299. This, however, can only give a general idea of the features of the elastic system.

For example, a highly cambered concave loading branch of the curve $y=f(P)$ is indicative of a reduction in the rigidity of the elastic system with an increase in load. In most cases this is due to a preload provided in certain joints, bearings, etc. When the load exceeds the magnitude of the preload, the joint opens and the rigidity of the system drops.

A convex loading branch of the same curve shows that the rigidity of the elastic system increases with the load. This occurs, for example, when the

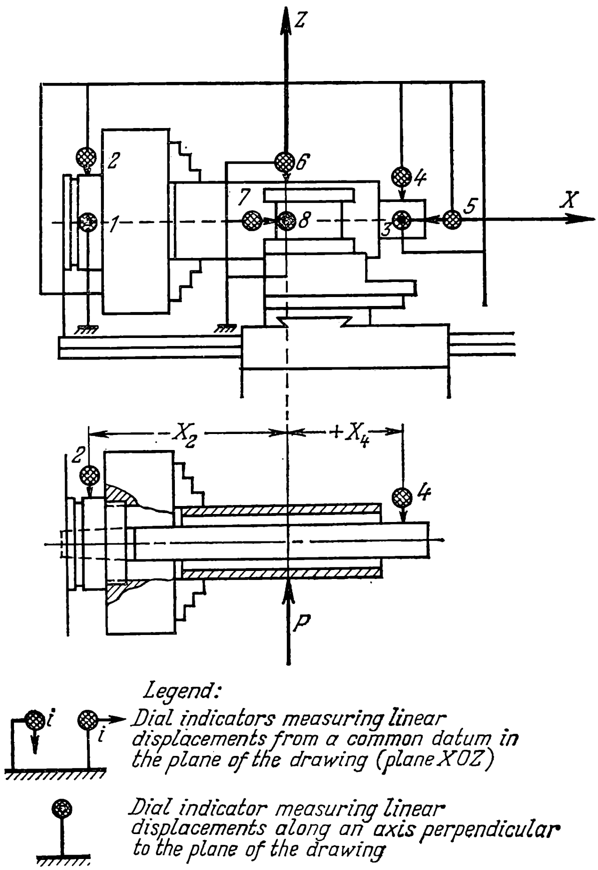


Fig. 300. Schematic diagram of instrument arrangement employed in drawing up a consolidated balance of elastic displacements in a lathe set up for holding the work in a chuck

joint surfaces have been poorly machined so that the mating surfaces contact only at certain points. Upon an increase in the load, the projections on the surface are deformed (flattened out) and the bearing surface increasas, as does the rigidity.

As a rule, the static characteristic of the elastic system in a machine tool, $y = f(P)$, is a nonlinear function with a hysteresis loop. Its nonlinearity may be due to the nonlinearity of the elastic forces or to the influence of the friction forces. In more profound machine tool research, it is of prime importance to separate these two types of forces since the roles they play in dynamic

processes differ in principle. With high-quality assembly and no clearance or weak members in the supporting system of the machine tool, the area of the hysteresis loop will be reduced and the characteristic itself will approximate a linear function. To simplify analysis, the characteristic is linearized in any case. Expressing such a linearized characteristic in analytic form we obtain the known relationships

$$k=\frac{x_{out}}{x_{in}}=\frac{y}{P} \text{ or } j=\frac{P}{y}$$

A knowledge of the overall rigidity is not always sufficient, however. Frequently, in order to avoid the advent of a construction with low rigidity because of a single weak element, it becomes necessary to assess the rigidity of separate elements or units, and the quality of their manufacture and assembly. Then, usually under laboratory conditions, the balance of elastic displacements is drawn up (or, more exactly, the structure of the displacements is determined). For this purpose, the machine tool is subjected to static loading that imitates the action of the cutting forces. The displacements of the elements of the machine tool elastic system are measured and then recalculated in reference to the point of force application. The balance of elastic displacements may be a consolidated balance of elastic displacements,

Consolidated Balance of the Elastic Displacements in a Lathe Setup with the Work Held in a Chuck

Elements whose displacement determines the displacement of the point of force application	Components of the total displacement of the point of force application		
	δ_x (along X-axis)	δ_y (along Y-axis)	δ_z (along Z-axis)
Headstock	$\delta_{x1}=\Delta_5$	$\delta_{y1}=\frac{\Delta_1X_3-\Delta_3X_1}{X_3-X_1}$	$\delta_{z1}=\frac{\Delta_2X_4-\Delta_4X_2}{X_4-X_2}$
Carriage	$\delta_{x2}=\Delta_7$	$\delta_{y2}=\Delta_8$	$\delta_{z2}=\Delta_6$
Total displacement of the point of force application	$\delta_x=-\delta_{x1}+\delta_{x2}$	$\delta_y=-\delta_{y1}+\delta_{y2}$	$\delta_z=\delta_{z1}-\delta_{z2}$

Note: Δ_i = true displacement (measured from the undeformed datum) at the point whose displacement is measured by dial indicator i; X_i is the co-ordinate of the point whose displacement is measured by dial indicator i.

Fig. 301. Table listing the treatment of the results of measurement

giving a general idea of the distribution of displacements among the various units; a detailed balance of the elastic displacements of a unit, or a detailed balance of the elastic displacements of the machine tool. In drawing up a balance of this type, the full displacement of the point of force application is expressed as a fraction determined both from the inherent deformation of the elements of the elastic system and from the contact deformation in the joints.

General methodological instructions are given in a work of ENIMS dealing with the drawing up of the balance of elastic displacements in the rigidity investigations of machine tools. Illustrated in Fig. 300, as an example, is the arrangement of instruments for drawing up a consolidated balance of the elastic displacements in a lathe for the case when the work is held in a chuck. The table listing the treatment of the results of measurement is given in Fig. 301. The loading devices used for investigating the structure of the displacements are similar to that shown in Fig. 297. Dial indicators, reading to 0.01 and 0.002 mm, respectively, and mounted on stands on a common undeformable datum, are used as measuring instruments. A spirit level with graduations of 0.02 mm per 1,000 mm is employed to measure angular displacements.

It is a more laborious job to draw up a detailed balance since this requires experimental determination of the contact deformations characterized by the mutual displacements of the contacting bodies in their joints.

CHAPTER 28

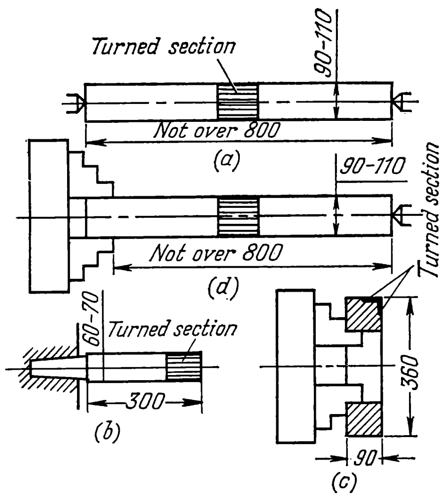
VIBRATION TESTS FOR MACHINE TOOLS. INSTRUMENTATION AND METHODS OF EXPERIMENTAL RESEARCH IN MACHINE TOOL VIBRATIONS

A loss of stability of the system is often observed in the operation of machine tools. This leads to the appearance of self-excited vibration which deteriorates the quality and accuracy of the machined surface, reduces the output and makes more frequent repairs necessary.

The instability of the system may be manifested in the form of nonuniform, stick-slip travel of the machine tool units, commonly observed at low sliding speeds. Such a travel, accompanying the cutting process, is especially harmful for machine tools used in finishing operations. In positioning motions, stick-slip phenomena reduce the accuracy with which tables, stanchions, columns and similar components are set to co-ordinate dimensions (positioning accuracy).

One of the main problems of present-day machine tool engineering is the development of machine tools with high vibration-proof properties. Previously (see Part Five, Chap. 12, Vol. 3), it was mentioned that the conditions for the loss of stability of a system are determined by the values of the parameters of all the elements of the closed dynamic system of the machine tool, as well as their linkages (primarily, the construction of the machine tool and its operating conditions). Defects in the manufacture and assembly may affect the values of the parameters of the elastic system, introduce new linkages into the system, lead to various kinds of disturbances, etc., and thus lower the established stability level of the system. Consequently, the stability of the dynamic system of a machine tool may serve as a reliable criterion of efficient design and the quality of manufacture and assembly of a machine tool. It can be used to assess the degree with which the performance of a machine tool complies with the design processing capacity. This is the basis for a number of special tests which are applied both to pilot models of new machine tools and to lot-produced models. The procedure for carrying out laboratory tests on the vibration-proof properties of pilot models of general-purpose lathes and knee-type milling machines was developed in ENIMS under the supervision of V. Kudinov.

Before applying these tests, the machine should be checked for accuracy and rigidity; installed on a foundation having a peak-to-peak amplitude of vibration due to external influence not over 5 or 10 microns; and levelled



Blank	v, m/min	s, mm/rev
a	20 to 150	0.1 to 1
b	20 to 150	0.1 to 0.3
c	20 to 120	0.12 to 0.3
d	20 to 150	0.1 to 1

Fig. 302. Kinds of blanks and the speeds and feeds recommended for lathe vibration tests under laboratory conditions (the dimensions and other data are shown for lathes with a maximum workpiece diameter of 400 mm)

to a spirit level by means of wedges or adjusting shoes. The machine is installed without tightening the foundation bolts if such are provided for.

We shall demonstrate the procedure for conducting such tests on the example of a lathe. The test includes the determination of the so-called limiting chip and its dependence on the cutting speed for typical kinds of machining and at a definite rate of feed. The limiting chip is understood to mean the maximum width of the uncut (undeformed) chip that can be removed without leading to vibration.

The main typical blanks used in lathe vibration tests are (Fig. 302): a shaft turned between centres (Fig. 302*a*) with a length 0.7 of the maximum length that can be turned; cantilever shaft secured in the taper hole of the spindle (Fig. 302*b*); a ring clamped by a chuck (Fig. 302*c*); and a shaft clamped in a chuck and supported by the tailstock centre (Fig. 302*d*).

The dimensions of the blanks shown in Fig. 302 are recommended for checking a lathe with a height of centres equal to 200 mm. The blanks are to be made of steel 45.

A saving of metal can be achieved by employing, for example, the built-up mandrel designed in the Orgstankinprom Process Engineering Institute (Moscow).

Here the section to be turned is on an interchangeable ring (Fig. 303). A straight-turning tool of definite geometry is recommended as the cutting tool. High-speed steel tools are used for cutting speeds $v \leqslant 30$ m per min, and carbide-tipped tools of grade BK8 or T15K6 cemented carbide for speeds $v > 30$ m per min. The size of the toolholder and the tool overhang are also specified. The tool should be precisely set in height to the tailstock centre. A close watch should be kept during the test on the condition of the cutting edge on the tool. Tools should be resharpened without deviation from the accepted geometry.

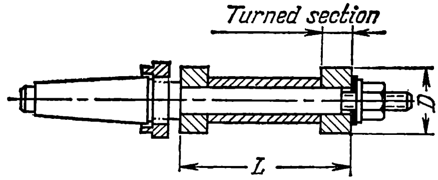


Fig. 303. Built-up test mandrel designed in the Orgstankinprom Process Engineering Institute

The tests are conducted in the recommended speed and feed ranges (listed in the table in Fig. 302 for lathes with a 200-mm height of centres) at all the speed steps of the main drive and at 3 to 5 different rates of feed.

Determination of the limiting chip is quite a difficult operation since the machine tool is very unstable at the limiting speeds and feeds. Therefore, in determining the limiting chip on the basis of the chatter marks on the machined surface (at $v \leqslant 25$ m per min), from the characteristic noise during machine operation ($v = 25$ to 50 m per min) or the heavy waviness and serrations of the chip being removed ($v = 50$ to 200 m per min), an error up to 1 mm or even more is possible.

The accuracy of limiting chip determination can be raised and more reliable results can be obtained by making simultaneous observations of the vibration of the workpiece, headstock housing, table, etc., in the course of the test by means of quick-response apparatus (such as a variable-reluctance pickup).

Figure 304 shows the arrangement of the apparatus for determining the maximum chip. The variable-reluctance pickup (Fig. 305) is a coil with an iron core connected into an a-c circuit. At constant voltage and frequency of the power supply, the current flowing through the coil is directly proportional to the gap between the core and armature. If some element of the machine tool elastic system is used as the armature (in Fig. 304 the blank is the armature) and the pickup is located so that the gap is from 0.1 to 0.3 mm, the displacement of the element can be noted.

Such pickups are employed in conjunction with various bridge circuits and a reference pickup; an a-c amplifier is connected into the diagonal of the bridge. The signal from the amplifier passes through a special co-ordinating attachment to a cathode-ray oscillograph. The moment the limiting chip is reached, a sharp increase in the amplitude of vibration will be observed on the screen of the oscillograph. Together with the determination of the limiting chip it proves expedient to note the frequency f of the vibration that appears at this moment on the machine tool.

The frequency f can be calculated from the wave length l mm and the known cutting speed v m per min. Thus

$$f = \frac{16.7\,v}{l} \text{ cps}$$

or it can be determined from the reading of the oscillograph screen. A knowledge of these frequencies (the so-called frequencies of the natural vibrations of unstable form) is useful in further investigations of the machine tool.

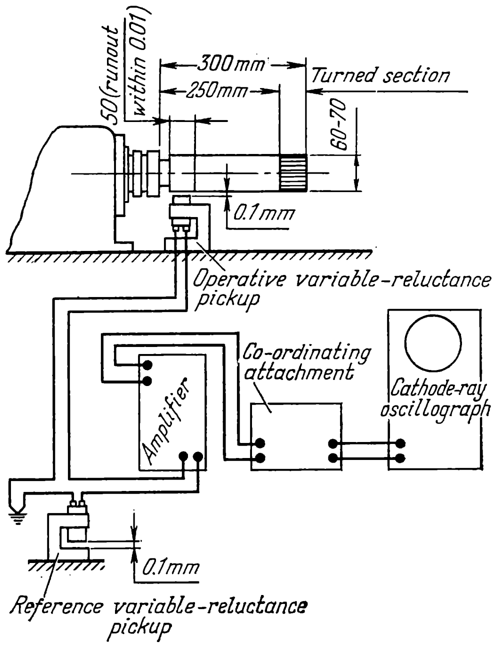


Fig. 304. Pickup arrangement and apparatus connections for determining the maximum chip

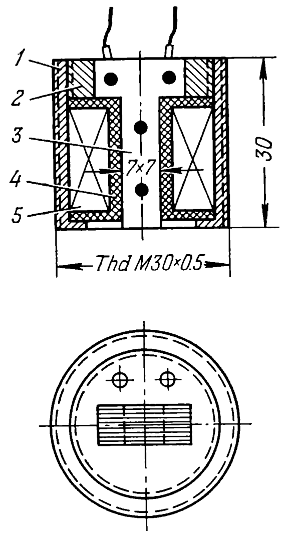


Fig. 305. Variable-reluctance pickup:

1—pickup case; *2*—nut; *3*—core; *4*—coil form; *5*—winding

The data obtained can be used to plot curves showing the dependence of the limiting chip on the cutting speed at various rates of feed for each kind of machining method—the so-called borderline of stability curves. Such borderline curves are shown in Fig. 306. The height of the broken lines—the borderlines of stability—above the axis of abscissas is an indication of the capacity of the machine to remove a certain chip without vibration. Borderlines of stability are also convenient in investigating the effect on the vibration-proof properties of various changes in construction, introduced in the process of testing the pilot model, or of errors in manufacture.

By way of example, we shall consider the influence of the magnitude of the radial clearance in the front spindle bearing of the model 1K62 engine lathe on the amplitude of the relative vibration of the work being turned. Different values of the radial clearance corresponded to different deflections of the spindle nose when the workpiece was loaded with a horizontal force of about 600 kgf.

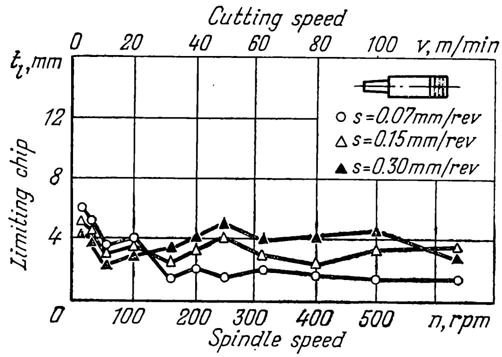


Fig. 306. Borderline of stability curves for the model 1K62 lathe in turning a cantilever blank

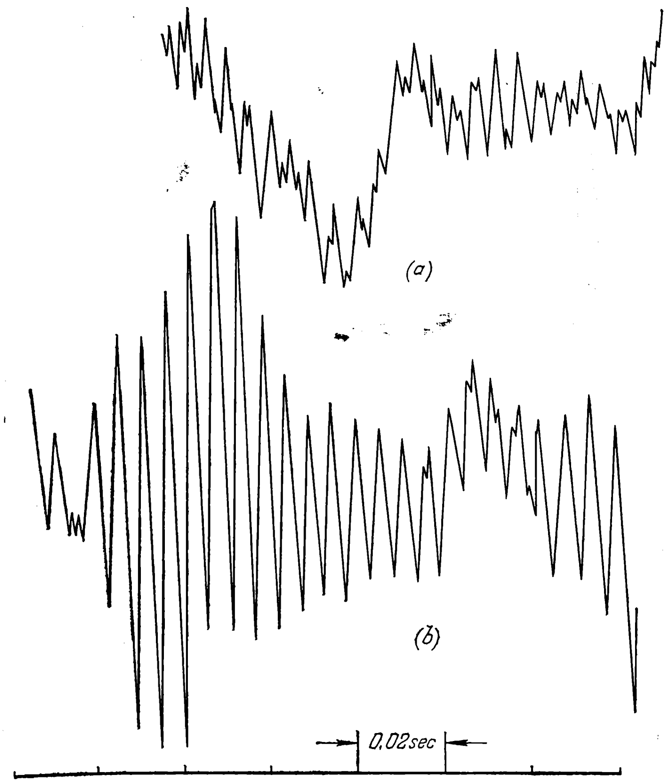


Fig. 307. Oscillograms of the relative vibrations of the workpiece being turned in the model 1K62 lathe

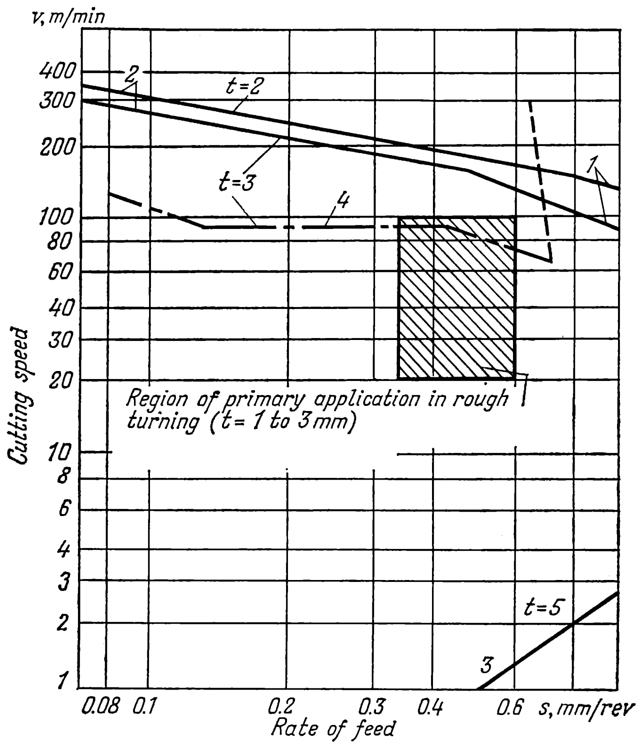


Fig. 308. Production characteristic of a lathe with the plotted borderline of stability

In Fig. 307*a*, with a deflection of 0.02 mm, the level of relative vibration was substantially less than in Fig. 307*b* when the deflection was 0.05 mm. In both cases, the spindle speed was $n = 500$ rpm and the depth of cut $t = 2$ mm. Obviously, at a deflection of 0.05 mm, the system has a lower margin of stability.

ENIMS recommends that the production characteristics of a machine tool be used to assess limitation in the processing capacity on the basis of the vibration-proof properties. As is known, the production characteristic of a machine tool, proposed by A. Kashirin, enables the field of practical application to be determined.

Production characteristics (Fig. 308) are cutting speed (recalculated to a diameter of 100 mm) vs feed curves, plotted in logarithmic co-ordinates, for the most typical kinds of machining and cutting conditions (speeds and feeds) of the machine tool.

The following borderlines of machine tool application are plotted on the production characteristic:
1—maximum effective power available on the spindle or available power of the drive motor, for example, for lathes

$$v = \frac{6,120 N_e}{P_z}$$

where $P_z = C_{P_z} t s^{0.75} v^{-0.15}$;
2—speed corresponding to specified tool life, for example, in machining steel 45 with a straight-turning tool tipped with grade T15K6 cemented carbide with a plan approach angle $\varphi = 45°$ and a life $T = 60$ min

$$v_T = \frac{165}{t^{0.15} s^{0.3}}$$

3—strength of the feed mechanism, using the formula

$$P_x = C_{P_x} t s^{0.5} v^{-0.4}$$

4—vibration-proof properties.

The last of these borderline curves is plotted on the basis of the experimentally obtained borderlines of stability (Fig. 306). The limiting chip values t_l from these curves are plotted on the production characteristic. Points with the same t_l value are connected together by broken lines. This forms a family of borderlines of machine tool application, in respect to the vibration-proof properties, for a definite depth of cut.

If, now, the region of primary application of the machine tool is plotted on the production characteristic (hatched area in Fig. 308) for various kinds of machining (roughing, finishing, etc.), on the basis of statistical data, it will become evident whether the machine satisfies the requirements or not.

The conditions and the speeds and feeds for shop tests are determined in the course of laboratory tests. From the various kinds of possible blanks, the one providing the least stability is chosen. As recommended feed. use is made of the average rate which leads to a sharp increase in the amplitude of vibration when the depth of cut is increased. The cutting speed is selected equal to that in the zone of the minimum value of the limiting chip on the borderline of stability curve for the previously selected rate of feed. The depth of cut should be slightly less or equal to that at the maximum chip obtained on most of the machine tools of the given model.

Shop tests of lot-produced machine tools are conducted according to a curtailed schedule. Their aim is to check the vibration-proof properties as a criterion of the quality of manufacture and assembly of each machine tool.

In the course of the shop tests, the blank is machined at the speeds, feeds and depth of cut selected as indicated above (for the model 1K62 lathe,

$v = 30$ to 35 m per min, $s = 0.12$ mm per revolution and $t = 2$ mm). The machine tool is accepted as vibration-proof if no traces of vibration (chatter marks) are found on the machined surface; otherwise it is rejected as lacking vibration-proof properties.

Under laboratory conditions, the vibration-proof properties of a machine tool can be determined, without cutting metal, by the gain-phase-frequency characteristic (GPFC) of the system, making use of the frequency stability criterion. This method proves useful in assessing the influence of any one parameter on the stability of the system. In this case the GPFC of a disconnected system is used (see Part Five, Chap. 12, Vol. 3); it is obtained by multiplying the GPFC for cutting by that of the elastic system. At the present time the GPFC of the cutting process is usually taken on the basis of available experimental data and in its simplest form, which is the coefficient of proportionality k_c between the cutting force and the variation in chip thickness (transfer factor). It is equal to the product of the specific cutting force k (for steel, $k \cong 20{,}000$ kgf per sq cm) and the width b of the chip.

The GPFC of the elastic system of a machine tool can be obtained experimentally. To this end, vibrations are excited in the system by means of a vibration generator, or simply vibrator, which produces a sinusoidal exciting force in the direction of the cutting force, in the frequency range of about 30 to 300 cps, between the carriage and the blank.

An electromagnetic vibrator (Fig. 309) can be employed in these experiments. The main part of this vibrator is the electromagnet on whose core *2*, coil *3* with two windings is tightly fitted. One of these windings (control winding *4*) is supplied with direct current and produces a constant magnetic flux in the core which attracts armature *1* of the electromagnet with a definite force. Alternating current of definite frequency is supplied to the second winding *5*. The result of the action of the two windings is a periodic variation of the force with which the armature is attracted to the core of the electromagnet.

The armature of the electromagnet is linked to holder *7* which is rigidly clamped in the carriage (in a lathe). The core of the vibrator is clamped on mandrel *8* which is held between centres or in a chuck between the headstock and tailstock. Therefore, the periodic force produced by the vibrator is transmitted to the elastic system of the machine tool.

The design of a vibrator usually incorporates means for changing the direction of the exciting force, this direction being selected in accordance with the direction of the resultant cutting force. The magnitude of the exciting force is recorded by means of a special built-in dynamometer *6* with a capacity pickup which is a variable-capacity condenser. The dynamometer is shown schematically in Fig. 310. The force is sensed by a rigid membrane, the upper part *1* of the dynamometer serving this purpose. One of the plates of capacitive pickup *2*, electrically insulated from the body, is linked

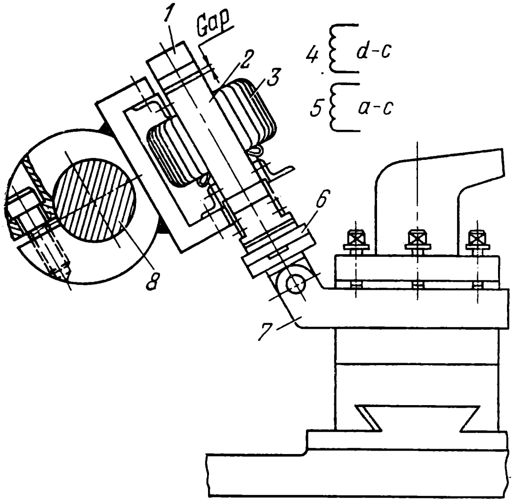


Fig. 309. Principle of the electromagnetic vibrator

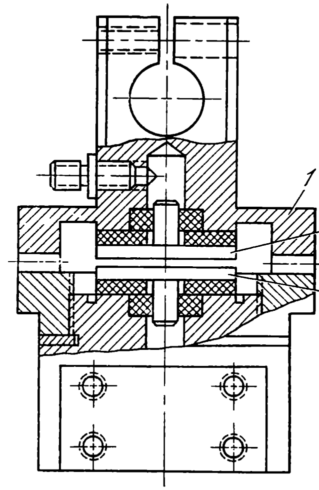


Fig. 310. Principle of the capacitive pickup for force measurements

to the centre of the membrane. The second plate *3*, also insulated from the body, is separated from the first plate by an air gap of 0.1 or 0.2 mm.

The construction of the pickup is such that the magnitude of the initial clearance in the gap can be adjusted. The variable force bends the membrane, varying the distance between the plates and, consequently, the capacity of the condenser.

The rigidity of the membrane is selected so that in operation the gap varies within limits up to 30 per cent of the initial value (within such limits the characteristic of the pickup can be assumed to be practically linear).

The pickup is usually connected into a bridge circuit whose measuring diagonal is connected through an amplifier with a loop oscillograph which records the force values on film.

Using a vibrator, the gain- and phase-frequency characteristics of the elastic system are recorded for various frequencies ω with the machine tool standing idle (see Fig. 313). To this end, the curve of the variation of force P (input co-ordinate of an element of the "elastic system") and the curve of the relative displacement y_2 of the blank and carriage normal to the surface being machined are simultaneously recorded on the film. The relative displacement y_2 can be conveniently recorded with the head of vibration transducer, type K61-A, having an elastic element on which wire resistance pickups are glued. The wire pickup (Fig. 311*a*) is a grid of thin, usually constantan, wire. The

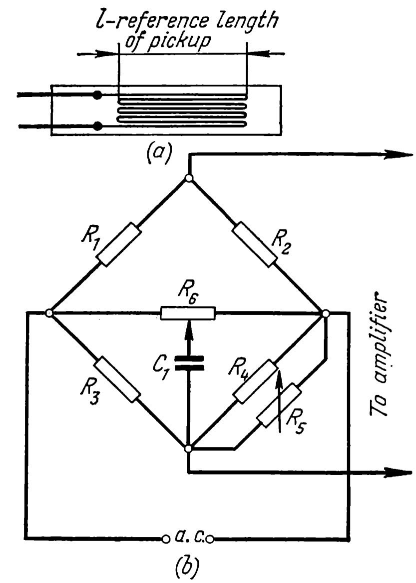


Fig. 311. Wire pickup (*a*) and the a-c bridge circuit (*b*) using wire pickups for deformation measurements

pickup is glued in a definite manner to the surface of the body being deformed. The deformation of the body leads to either extension or contraction of the grid wires, accompanied by a change in the ohmic resistance of the pickup which is registered by electrical means. The pickups are usually connected into a bridge circuit to increase the sensitivity of measurement. One of the commonly employed bridge circuits for operation with an amplifier is shown in Fig. 311*b*. The working pickups R_1, R_2, R_3 and R_4 are connected into the arms of the bridge which is supplied by alternating current. The current frequency (1,000 to 10,000 cps) should be deliberately higher than the frequency of the process being recorded. The effective resistances of the bridge arms are balanced by means of potentiometer R_5. The reactances of the arms (the bridge has an a-c supply) are balanced by potentiometer R_6 and variable-capacity condenser C_1, connected in parallel with the arms R_3 and R_4.

The bridge output (measuring diagonal) is connected to a special amplifier. There is no current in the diagonal of the bridge when it is balanced. A change in the resistance of the pickups unbalances the bridge and the resulting current is the larger, the greater the change in the resistance of the pickups. By sending the signal obtained from the bridge to the amplifier and further to the oscillograph, it is possible to record very slight deformation of the elastic element in the head of the vibration transducer. The elastic element of the type K61-A transducer head (Fig. 312) consists of spring *1*, bent into an arc, and needle *3*. The ends of the spring are brazed to strip *4*. Pickups *2* are glued in pairs on the inside and outside of the spring, near the needle. To record relative vibration, the head is clamped on the carriage and the needle contacts the surface of the mandrel whose relative vibration is being investigated. Shown in Fig. 313*a* is a sample film

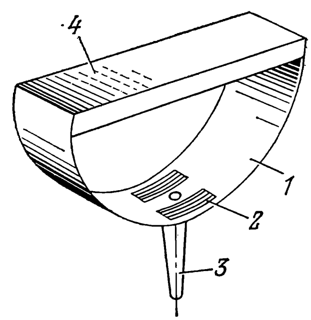


Fig. 312. Elastic element of the vibration transducer head, type K61-A, designed by S. Kedrov of ENIMS

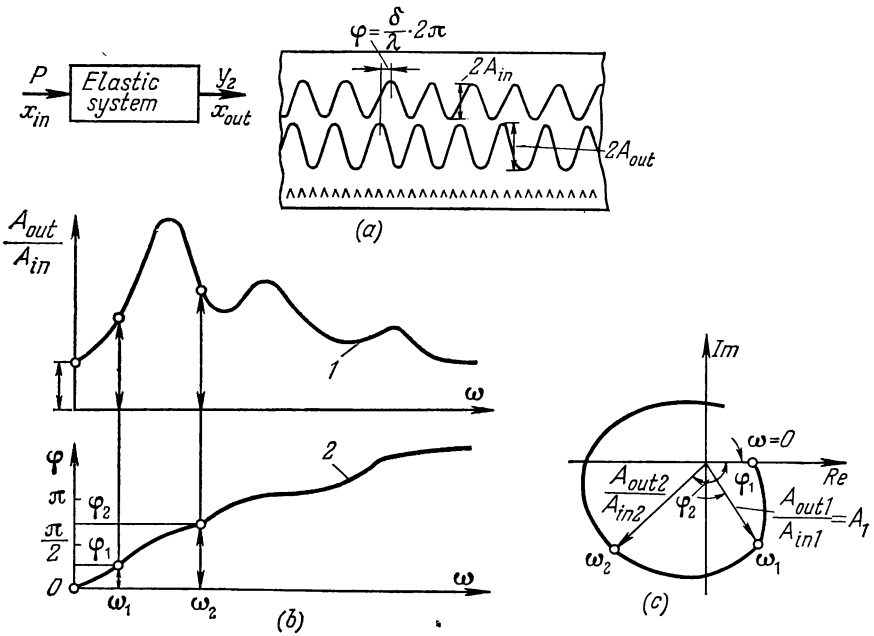


Fig. 313. Plotting the gain-phase-frequency characteristic (GPFC) from experimental data:

(*a*) sample film frame; (*b*) gain- (*1*) and phase- (*2*) frequency characteristics of the elastic system; (*c*) the GPFC of the elastic system

frame with recorded curves representing the vibration in force P and the relative displacement y_2 for a definite frequency. Processing the data on the film, the gain- and phase-frequency characteristics are plotted (Fig. 313*b*) and, on their basis, the gain-phase-frequency characteristic of the elastic system (Fig. 313*c*) in the plane of the complex variable.

To obtain the phase φ from the data on the film, the ratio of the wave displacement δ of the two vibrations to the wave length λ is determined. This ratio, multiplied by 360° or by 2π, is the phase in degrees or radians. Thus

$$\varphi=\frac{\delta}{\lambda}\,360°$$

The multiplication of the GPFC's of the cutting process and of the elastic system consists practically in scaling the experimentally plotted GPFC of the elastic system k_c-fold. The appearance of the GPFC of the disconnected system (Fig. 314), obtained by multiplication, is the basis for assessing the stability of the closed system which represents the machine tool in operation.

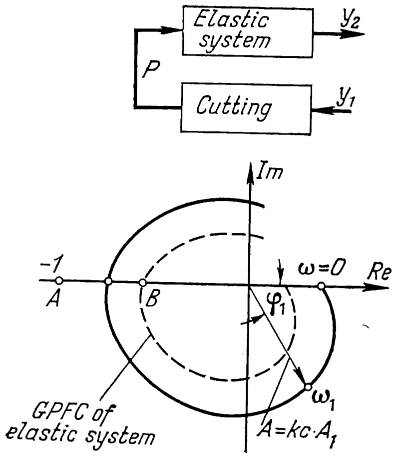


Fig. 314. Disconnected system and its gain-phase-frequency characteristic

It can also be used to find the limiting value of coefficient k_c (intercept AB in Fig. 314) and, consequently, the maximum chip width b at which the system remains stable.

The influence of the nonlinearity of the elastic system of machine tools and the complexity or unavailability of the necessary apparatus restrict the use of the frequency method in many cases. On the other hand, the assessment of vibration-proof properties on the basis of the GPFC does not require the conversion of metal into chips in the tests. This method also proves useful in assessing the influence of any one parameter of the system (rigidity, mass, cutting speed, etc.) on its stability and in checking the results of theoretical calculations in machine tool stability.

If a machine tool is not sufficiently vibration-proof, one method of revealing the cause of this is to analyse the form of the vibration of its elastic system.

The form of the vibration of a linear system is the whole complex of ratios of the displacements of various vibrating points to the displacement of any single point of the elastic system, determined at a definite moment of time (taking phase shift into consideration) for a given frequency.

The form of the vibration of a nonlinear system, such as a machine tool, can be determined only very approximately, giving, however, an idea of the displacements of certain points of the machine tool in vibration.

The form of the vibration is determined for all frequencies of intensive vibration which were revealed in conducting the vibration test by cutting metal. Vibration is excited for this purpose either by cutting metal at vibrational speeds and feeds or by means of a vibrator, as in plotting the GPFC of the elastic system.

Before selecting the points for measuring the vibration, the intensity of vibration is roughly estimated. At a place sufficiently convenient for mounting a pickup and where the vibration is very intensive and of regular form, the control, or reference, point pickup is installed for the duration of the experiment. This can be a pickup for measuring relative vibration if it is mounted on an undeformable datum. In a lathe, for example, the vibration of the control point (vibration of the workpiece in a vertical plane at the tailstock) is measured by a variable-reluctance pickup mounted on the bed which is practically not deformed in a vertical plane during vibration. The

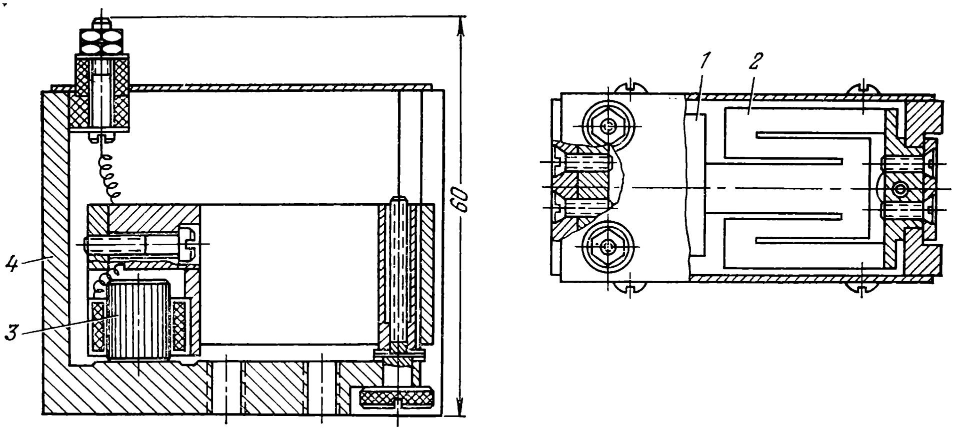


Fig. 315. Seismic-type vibrometer with a variable-reluctance pickup designed by S. Kedrov of ENIMS

vibration at the control point can be measured with a vibrometer of the seismic type as well. One such vibrometer with a variable-reluctance pickup is illustrated in Fig. 315. The vibrometer is secured at the control point. At this its body *4* begins to vibrate, repeating the vibration of the part on which it is mounted, and seismic mass *1*, suspended from two very flexible parallel flat springs, or reeds, *2*, remains practically stationary (under the condition that the natural frequency of vibration of the mass on the springs is substantially lower than the frequency of vibration of the body). The absolute vibration of the body, in reference to the stationary seismic mass, is recorded by means of variable-reluctance pickup *3* which responds to variation in the gap during vibration.

Other points of vibration measurement are chosen so that it will be possible to reveal the nature of the deformation or displacement of all the parts making up the elastic system of the machine tool. Parts with a comparatively small amplitude of vibration can be assumed to be rigid. Prior to determining the form of vibration, a diagram is drawn showing the location of all the points of measurement on the machine, and the points are numbered (points *1* through *42* in Fig. 316).

The absolute vibration of the elastic system can be conveniently measured with a type K61-A vibration transducer, having wire pickups, invented by S. Kedrov of ENIMS. The head of the transducer was previously described and is illustrated in Fig. 312. This head is secured to a heavy holder or body which is held in the hand during measurement, applying the needle of the sensitive element to the vibrating object with a definite pressure. This

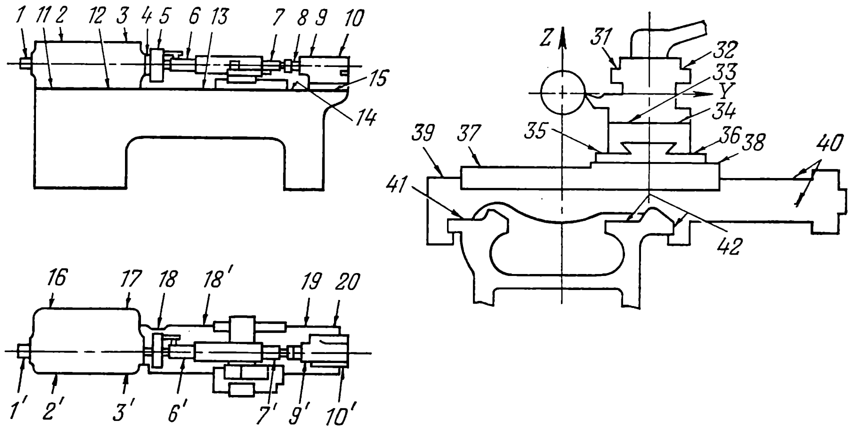


Fig. 316. Typical diagram showing the location of the points of measurement for determining the form of vibrations of a lathe

pressure is checked by a microammeter (50-0-50 microamperes), built into the body of the transducer and connected in parallel with the loop of the oscillograph (Fig. 317). Setting the switch to the COARSE position, the pressure is increased until the microammeter hand reaches zero. After this, by

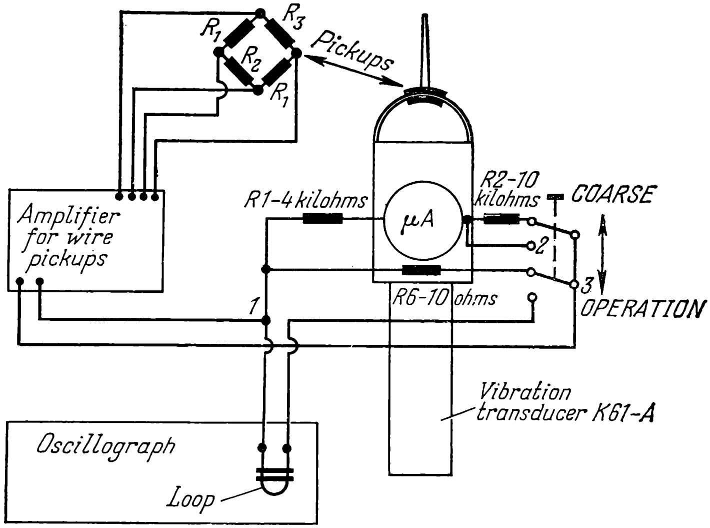


Fig. 317. Principle of the vibration transducer, type K61-A

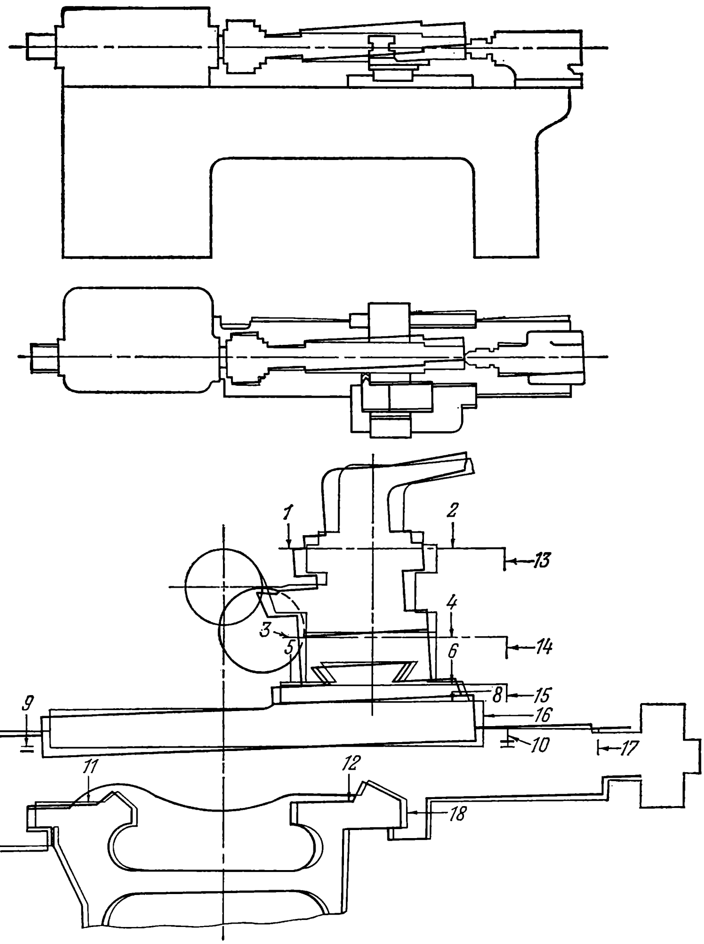


Fig. 318. Form of vibrations of the model 1Д62M engine lathe in cutting metal (without torsional vibration)

switching over to OPERATION, the measuring diagonal of the bridge is connected through the amplifier to the loop. This also cuts the resistor 2-10 kilohms out of the microammeter circuit enabling the pressure to be more finely adjusted. The shunting effect of the circuit, *1*—resistor 1-4 kilohms—microammeter (having a high internal resistance)—*2*—*3*, on the loop is negligible.

The main tone of the vibration of the spring (500 cps) is suppressed if the elastic element of the transducer is applied with sufficient pressure. Rocking of the operator's hand holding the transducer is practically insignificant for a short film frame.

After processing the oscillogram data, the results of measurement (control point displacement, points of measurement and the phase between them) are set down in a test record sheet. The obtained data (an average of 3 to 5 measurements for each point) enable the relative amplitudes of vibration of the points to be found, taking the phase into consideration. Then, in a scale convenient for use, the obtained form of the vibrations can be plotted on the schematic drawing of the machine tool (Fig. 318). Points (*1* through *18*) of vibration measurement are shown on the schematic drawing of the carriage.

Along with the recording of the translatory displacements of the assigned points, the form of the torsional vibration of the drive is determined. This can be achieved with the aid of wire pickups glued on the shafts of the drive as in the arrangement for torque measurements (see Fig. 350).

Relative motions of the workpiece and cutting tool, originating self-excited vibration in cutting, are due to the displacement of certain elements of the machine tool. The form of the vibrations clearly demonstrates which elements are involved, and to what extent their displacement influences such vibration. In Fig. 318, for example, it is evident that in a lathe the greatest displacement is observed at the rear end of the workpiece. The carriage has a displacement only about one-fifth as large. A knowledge of the form of the vibration is of prime importance in substantiating the design diagram of the elastic system of a machine tool being designed (see Part Five, Chap. 12, Vol. 3).

CHAPTER 29

IDLE-RUN VIBRATION INVESTIGATIONS IN MACHINE TOOLS

One of the conditions ensuring a high class of dynamic performance of a machine tool is the lowest possible level of relative vibrations of the cutting tool and workpiece in operation.

These vibrations may be either self-excited or forced.

The procedure for determining the limiting conditions (cutting speed and feed, chip cross section, etc.) at which self-excited vibrations appear was described previously. This chapter will deal with forced vibrations.

Among the factors that intensify the relative vibrations of the cutting tool and workpiece are various external forces and kinematic disturbances which, in a stable system, have a detrimental effect on the quality of surface finish and are one of the sources of increased noise in machine tools. These disturbances include periodic forces due to unbalance of electric motor rotors, grinding wheels, pulleys, blanks, etc.; periodic forces due to errors in toothed gearing (for instance, pitch errors may be manifested with a frequency of $\frac{nz}{60}$, where n and z are the speed, rpm, and the number of teeth of the gear, respectively), errors in and the influence of belt drives (for example, with a frequency equal to that of the transverse vibration of the sides of the belt), errors of spline and key joints, coupling misalignment, waviness on the surface of antifriction bearing races (cases of resonance are possible if one of the forced frequencies coincides with the natural frequency of one of the bearing rings, in connection with which the nature of the fit of the bearing in the housing or on the shaft, and the preload value become extremely important), etc.

All these disturbances, in conjunction with the action on the machine foundation of external sources and changes in the working conditions of the machine tool, lead to forced vibrations which are evident in the form of waviness and microirregularities on the machined surfaces of the workpiece. The greatest effect of forced vibration is on the operation of finishing machines.

The effects of such periodic and nonperiodic disturbances, due to defects in the manufacture of the machine tool and, primarily, its drive, are especially pronounced in an idle run. Therefore, the level of vibration during an idle run can be conveniently used to assess the quality of manufacture and assembly. If two machine tools have the same frequency spectrum of

idle-run vibrations, the errors will be larger in the machine with the larger amplitude of vibrations.

Beginning with 1960, investigations have been conducted in ENIMS under the supervision of V. Kudinov and T. Vorobyova with the aim of developing an industry standard on the permissible idle-run vibration of machine tools.

These standards are to be based on the results of the statistical treatment of data on the measurement of the level of idle-run vibration of machine tools that have successfully passed accuracy, rigidity and vibration tests.

Of chief interest are the measurement of the relative vibrations of the cutting tool and the workpiece, since these directly affect machining accuracy, and a study of the effect of various kinds of disturbances on the level of these vibrations.

Since the disturbances subject to investigation are those due to defects of the machine tool or its component units, any other disturbances (for instance, those transmitted through the foundation) should have a considerably lower level. In some cases, after suitably checking the amplitude of vibration of the foundation, it may be necessary to provide supplementary vibration isolation. Such isolation is not usually required for standard-accuracy machine tools.

Since the vibration is measured with the machine tool running at various speeds, the readings of the instruments are strongly affected by the runout of the mandrel, held in the taper hole of the spindle and acting as the workpiece. ENIMS recommends the use of a special adjustable mandrel with a runout within 5 microns for lathe investigations, and a mandrel with lapped centre-holes and a balanced driving dog for grinder investigations.

The relative vibration of the mandrel is measured, as a rule, by means of the head of the type К61-А vibration transducer or an induction transducer, type ЛДС-13, which is mounted and clamped in the tool post or square turret (in the case of a lathe) or on the table (for a cylindrical grinder). Through a pad of aluminium or brass foil, several hundredths of a millimetre thick, the transducer needle (Fig. 319) contacts the cylindrical portion of the rotating mandrel. The pad reduces the wear of the needle, and the graphite lubricant applied to the surface of the mandrel prevents rapid wear of the foil. Neither the lubricant nor the foil introduces any appreciable distortion in the recorded curves. The relative vibrations of the table and wheelhead are measured additionally in cylindrical grinder investigations. In making measurements with a vibration transducer, its signal passes through an amplifier to the loop oscillograph where it is recorded on a film. The curves obtained are often of a nonharmonic nature, and the exciter of forced vibration can be revealed only by the consecutive disconnection of various elements of the drive or by changing the character of their influence. The results thus obtained are recorded on film and analysed.

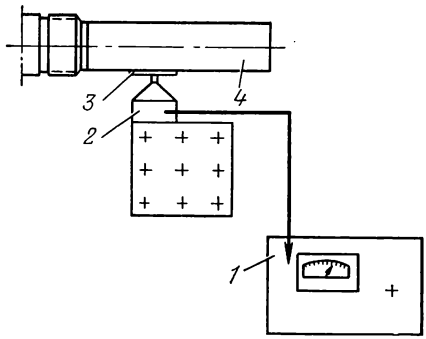


Fig. 319. Setup for measuring the level of relative idle-run vibrations of the workpiece:

1—instrument, type AB-1 (or ИВ-2); 2—induction transducer, type ЛДС-13; 3—pad of foil; 4—mandrel

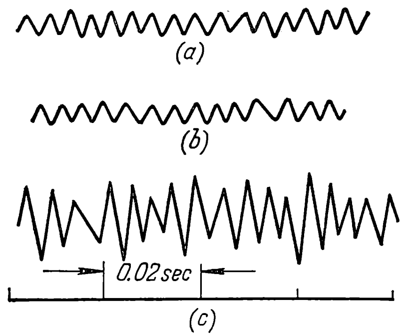


Fig. 320. Oscillograms of the relative vibrations of the cutting tool and workpiece in the model 1K62 engine lathe:

(a) with the drive belts removed; (b) with normally tensioned drive belts; (c) with heavily tensioned drive belts

The oscillograms in Fig. 320 show the effect of vibrations of the electric motor on the level of relative vibrations of a lathe at various degrees of tension of the drive belts. It is evident that with the belts normally tensioned the relative vibration is almost the same as when there are no belts, while heavy tensioning leads to a sharp increase in the amplitude of the vibration (3- or 4-fold), the frequency being also changed to some extent.

Induction transducer, type ЛДС-13 (Fig. 321), developed in ENIMS, can be efficiently applied for measuring relative vibrations. The transducer case, steel cylinder *7* which is 50 mm in diameter and 110 mm long, is mounted on the carriage or table of the machine. By means of screw *8*, needle *9* is linked to two coils *6* which are suspended on flat springs *1* having arc-shaped openings. When the needle is depressed, coils *6* are displaced in the annular gaps formed by pole shoes *2* of permanent magnet *3* on the inside, and poles *5*, concentric with the pole shoes, on the outside. The permanent magnet is press-fitted into silumin sleeve *4* in the centre of the transducer. The coils are connected in series and opposed to each other, and their leads are brought out of the casing. Upon displacement of the coils in the annular gaps, a current is generated in them which has a voltage proportional to the velocity of displacement. The induction transducer, being a powerful generating pickup, requires no special amplifier as do the wire pickups. This is especially valuable under shop conditions.

The ЛДС-13 transducer can operate together with the electronic instrument, type AB-1 (designed in ENIMS), and is used for vibration analysis. The circuit of the instrument includes an integrating cell enabling the ampli-

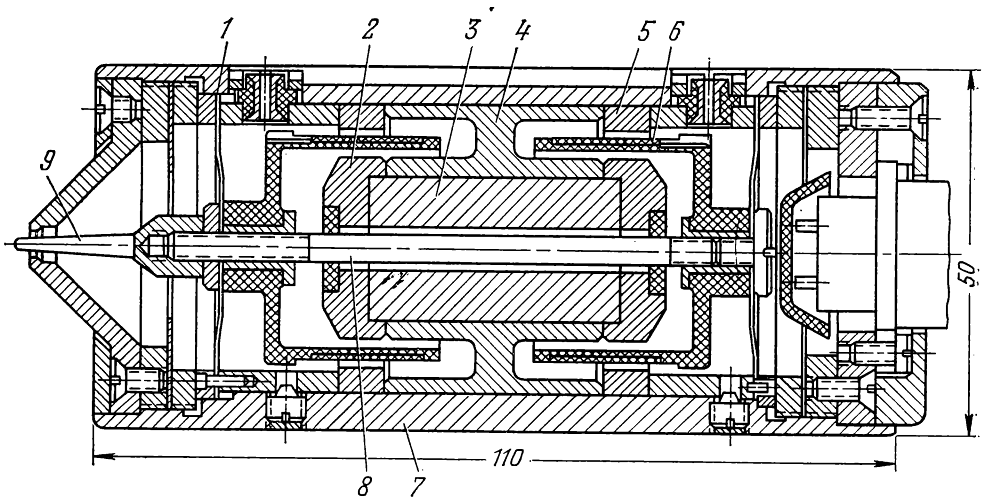


Fig. 321. Induction transducer, type ЛДС-13 (designed in ENIMS), for measuring the velocity of relative displacements

tude of any component of the relative vibration to be directly determined by the deflection of the hand of a microammeter in the instrument. The analyser is sufficiently portable and is convenient for conducting a rapid analysis of a complex vibration to determine the levels of its separate frequency components from the readings of the indicating instrument. If the levels of the components of a complex vibration are the same, the instrument can single out vibrations differing in frequency by at least 5 per cent.

Illustrated in Fig. 322 is a typical vibration frequency spectrum of an idle run of the model 1К62 lathe, plotted with the aid of the AB-1 instrument. The frequency spectrum of idle-run vibration is dictated mainly by the most powerful exciters and the natural vibration of the elastic elements of the system. To check the quality of machine tool manufacture, ENIMS proposed that norms be established for the level of idle-run vibration in the frequency ranges.

This proposal is based on theoretical principles developed by V. Kudinov concerning the relation between the dynamic machining errors upon external force or kinematic disturbances and the amplitude A_{ir} of the relative vibrations of the cutting tool and workpiece in an idle run.

According to these principles, the amplitude A_{ir} of forced idle-run vibrations is increased in cutting if the length of the vector A_{va} of the gain-phase characteristic of the disconnected system for the frequencies of these vibrations is less than unity (Fig. 322). The increase will be the greater, the closer

these frequencies are to the natural frequencies of unstable forms of vibration of the system and the lower is the margin of stability of the system. On the contrary, the amplitudes A_{ir} of relative vibrations are reduced if for their frequencies the value of coefficient $A_{va} > 1$.

As a rule, the amplitude of vibrations whose frequency is a multiple of the spindle speed is considerably larger in an idle run than the amplitude of vibration of the unstable form. On the other hand, in machining at high cutting speeds and feeds, the amplitude of vibration of the unstable form increases up to the origination of self-excited vibrations. In this case, the amplitude of vibration of the unstable form can by far exceed the amplitude of vibrations of lower frequencies.

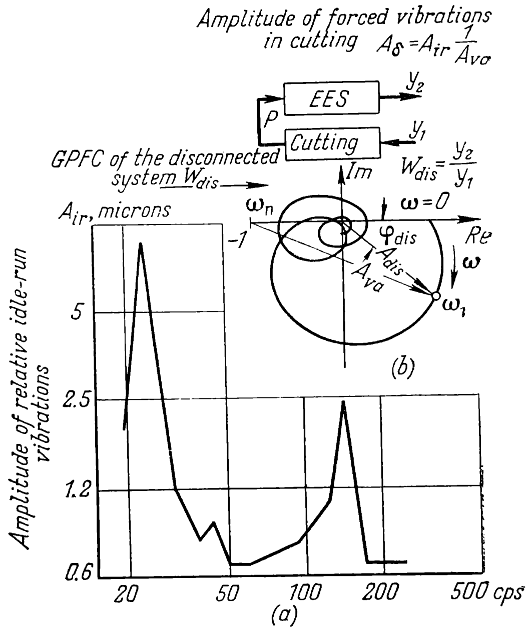


Fig. 322. Frequency spectrum of the idle-run vibrations of the reference lathe, model 1K62, at a spindle speed $n_{sp} = 1{,}250$ rpm and mandrel runout of 0.010 mm (*a*); determining the amplitudes of forced vibrations in cutting (*b*)

It follows from the above that, in the first place, it is very important to determine the natural frequencies of the unstable forms of vibration for various models of machine tools (for instance, in vibration tests involving metal cutting). In the second place, the norms for the level of idle-run vibrations for components, whose frequencies are located in the spectrum closer to possible self-excited vibrations in the system, should be much stricter than for other frequency ranges. This entails differentiation of the norms for the various frequency ranges.

An electronic vibration measuring instrument, type ИВ-2, was developed in ENIMS, for operation in conjunction with the ЛДС-13 transducer, with the aim of rapidly checking the level of idle-run vibration of machine tools within the frequency ranges. By means of a switch, this instrument can be set up to pass a given range of frequencies while signals corresponding to vibrations whose frequencies are not within this range are weakened. Like the AB-1 analyser the ИВ-2 instrument has an integrating cell enabling the average amplitude values of the relative vibrations to be determined in

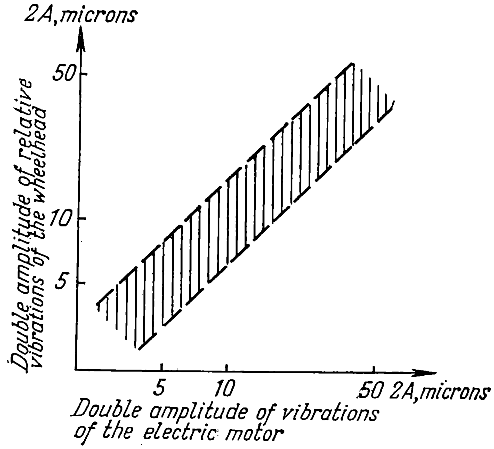


Fig. 323. Dependence of the relative vibrations of the wheelhead on the model 3110M grinder on the level of vibrations of the electric motor (after ENIMS)

microns, in the given frequency range, directly from the readings of the microammeter.

It can be seen from the spectrum of relative idle-run vibrations that very high vibration peaks are likewise observed for frequencies that are comparatively far from the dangerous ones (in the sense that the vibrations are intensified in cutting). As a rule, the highest of these peaks is the one with a frequency dictated by the speed of the electric motor (25 cps at $n_e = 1{,}500$ rpm). Even if the level of these vibrations is lowered, as is the case in cutting, their amplitude may turn out to be impermissibly large. For this reason, ENIMS proposed the additional measurement and establishment of norms for the vibration level of the most probable and powerful sources of external disturbances and, in particular, of the electric motor unit.

When the interrelation is found between the level of absolute vibrations of the electric motor for the machine tool being tested and the response of the system to these vibrations—in the form of a rise of the level of relative vibrations of the cutting tool and workpiece (a sort of transfer factor)—it becomes possible to check and establish norms for the vibrations of the electric motor itself, even before the motor is installed on the machine tool.

The vibration of the electric motor can be measured by means of the seismic induction transducer, type ЛДС-13С, whose design is based on type ЛДС-13. Following comparatively simple modification (consisting in removing the needle, increasing the flexibility of springs *1*, shown in Fig. 321, increasing the mass suspended from these springs and changing the end piece of the case), the ЛДС-13С transducer, in conjunction with the same recording instruments, enabled the level of the absolute variations to be comparatively easily measured at the points of interest.

As an example showing the application of the above-described instruments, certain data can be cited on the vibrations of electric motors and their influence on the level of relative vibrations in the machining zone. These data were obtained in ENIMS in tests of model 3110M cylindrical grinders.

An electric motor, one of the most probable sources of disturbances, vibrates in horizontal and vertical planes. The amplitude is large in the horizontal plane and reaches from 6 to 15 microns and, in particular cases, it is as high as 25 microns (without a belt drive). Especially marked in the spectrum of motor vibrations are the components with a frequency proportional to the motor speed (the main component due to unbalance of the rotor) and with a frequency of 100 cps (due to the lack of symmetry of the electromagnetic field). A belt drive usually raises the level of vibrations of the electric motor by 20 to 30 per cent in the direction of the drive. Cylindrical grinder investigations, conducted in ENIMS, have shown that the vibrations of the electric motor greatly affect the level of relative vibrations of the table and wheelhead housing. The magnitudes of the vibrations are in a probability relationship (Fig. 323). This is an indication of a pronounced positive correlation between the measured values.

The research carried out in ENIMS on a great number of machine tools, with the aid of these instruments, confirmed the main theoretical principles forming the basis for a study of idle-run vibrations of machine tools. The vibration-proof properties turned out to be low in all machine tools, without exception, in which a high level of vibrations was observed in an idle run, and especially at frequencies approaching those of natural unstable forms of vibrations. However not all machine tools that passed the tests on the level of idle-run vibrations, turned out to be sufficiently vibration-proof in cutting metal. This not only confirms the necessity for conducting such tests on each machine tool to more fully assess the quality of its manufacture and assembly, but also proves that idle-run tests in no way exclude the necessity for conducting vibration tests (by cutting or by using the GPFC).

CHAPTER 30

NOISE TESTS FOR MACHINE TOOLS

Stricter requirements are made to the manufacture of up-to-date machine tools that run at higher working speeds. A major criterion of the manufacture and assembly of machine tools is their noiseless operation. Analysis of the frequency spectra of machine tool noises enables the most intensive sources of noise to be revealed. Primarily, these are forced vibrations caused by manufacturing errors. Certain component frequencies are caused as well by self-excited vibrational processes and free vibrations with a frequency equal to the natural vibration of various components of the machine tool. The hydraulic systems and the electric motors are also intensive sources of noise. Most unpleasant in the motor is the high-frequency component of the magnetic noise emitted by the surface of the stator.

A high noise level is harmful to people working in the vicinity of a machine tool. The noise level is stipulated by regulations in production premises to ensure normal working conditions. Depending upon the frequency of the

TABLE 8

Class of noise	Noise characteristic	Permissible level, decibels
I	Low-frequency—highest levels in the spectrum are below a frequency of 350 cps	90
II	Medium-frequency—highest levels in the spectrum are below a frequency of 800 cps	85
III	High-frequency—highest levels in the spectrum are at frequencies, cps, of:	
	800 to 1,600	80
	Over 1,600 to 4,000	75

Note: The noise level in decibels is the Brigg's logarithm of the ratio of the power of the sound wave being measured to the power of the sound at the threshold of audibility at a frequency of 1,000 cps (10^{-16} W/cm^2).

noise, the general noise level in production premises is established on the basis of the data indicated in Table 8.

The noise of each separate machine tool should be considerably below the level indicated in the table and should be restricted by the limiting maximum levels of the frequencies making up the spectrum (Fig. 324).

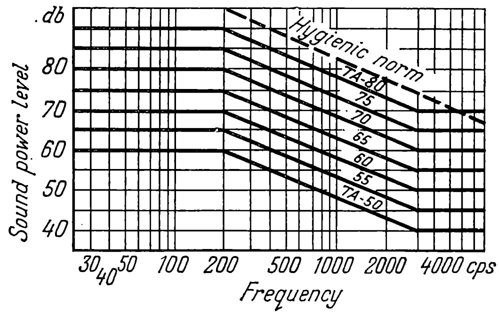


Fig. 324. Limiting maximum levels of component frequencies of the noise spectrum. *TA*—threshold of audibility

The selection of the values of the levels is governed, in the main, by the power of the main drive of the machine tool. The frequency spectrum of the noise should be within the boundary of the corresponding limiting line in the chart of Fig. 324, as follows:

Power rating of drive, kW	**Limiting line**
Up to 1	TA-50
1 to 1.6	TA-55
1.6 to 2.5	TA-60
2.5 to 4	TA-65
4 to 6.3	TA-70
6.3 to 10	TA-75
Over 10	TA-80

The noise is usually measured by objective noise level meters, types Ш-3, Ш-52 and Ш-60 (USSR), as well as decibel meters made by Dawe Instruments, Ltd (Great Britain) or by Bruel & Kjaer (Denmark). An objective noise level meter consists of a microphone, vacuum-tube amplifier with a rectifier and an indicating instrument graduated in decibels. The human ear is most sensitive to sounds of medium frequency (approximately from 700 to 5,000 cps). Consequently, the characteristics of the amplifier are selected to suit hearing sensitivity curves, the so-called equal loudness curves (Fig. 325). In this case, sounds which are equal in intensity but different in frequency are differently amplified and cause different deflections of the hand on the instrument.

A method of conducting noise tests for machine tools has been worked out by V. Vasilyev of ENIMS. The measurement of the general noise level is mandatory in testing pilot models of new machine tools and is selective for lot-produced models.

Consideration should be given to the following two circumstances in order to obtain correct results in noise measurements:

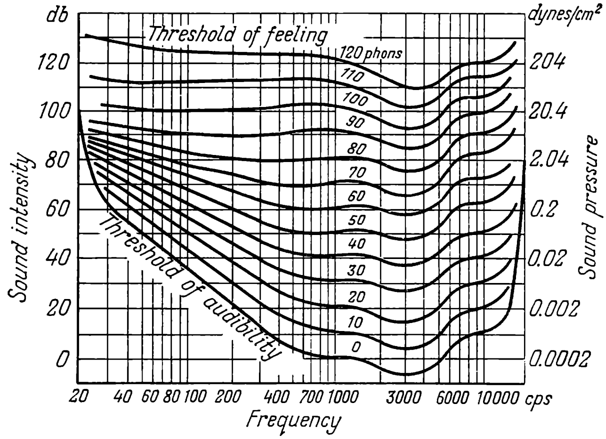


Fig. 325. Equal loudness curves

1. Noise disturbances from extraneous sources should be weaker than the minimum overall noise level by from 10 to 12 decibels. Otherwise, the results of the measurements must be corrected in accordance with the following data:

Difference in noise levels with the tested machine switched on and off, decibels . . .	1	2	3	4	5	6	7	8	9	10
Correction to be subtracted from the overall noise level, decibels	6.9	4.3	3	2.2	1.7	1.3	1	0.7	0.6	0.5

2. There should be no standing waves in the premises. If there are, the microphones should not be installed at the dead spots, which are points with a sharp drop in the noise level.

The machine tool is to be installed on elastic vibration isolators to prevent the transmission of vibrations from the machine to the foundation. The microphone of the objective noise level meter is set at the workplace (operator's position) about 0.5 m from the machine, and sometimes at several points at an equal distance around the machine (points *1*, *2*, *3* and *4* in Fig. 326). The noise level is measured in an idle run, consecutively engaging all the spindle speeds. The results of measurement are set down in the test record sheet and are used to plot curves of noise level vs spindle speed (Fig. 326). If the machine tool contains any units or installations that are independent sources of intensive noise, additional noise level measurements are made in the vicinity of these sources.

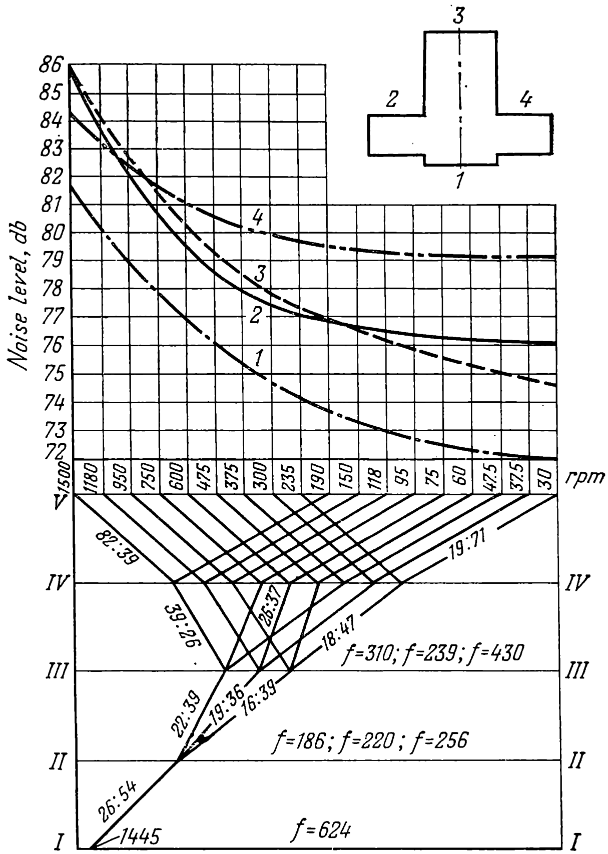


Fig. 326. Noise intensity level curves for various spindle speed steps

To reveal the causes of excessive noise during operation and possible methods for noise reduction, a frequency analysis of the noise is made during the testing of a pilot model at the spindle speeds at which the noise level was highest. Various types of sound frequency analysers can be connected to the noise level meter for this purpose. Analysers with band filters, for instance, are based on a set of electric filters, each passing vibrations only in a narrow frequency band and rejecting the others. By adjusting such an analyser to various frequencies, from the lowest to the highest, the noise level is determined, by means of an output instrument, in the various frequency bands singled out of the general spectrum of the noise being measured.

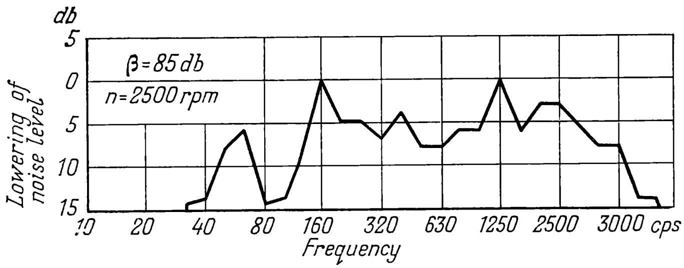


Fig. 327. Noise frequency spectrum

A condition ensuring accuracy of this analysis is the application of filters with a sufficiently narrow frequency pass band (a band width of one octave and a ratio of the mean frequencies of the adjacent bands of 1 : 2, or a band width of $\frac{1}{3}$ octave and a ratio of 1 : 1.26).

Noise spectrum curves (Fig. 327) are plotted from the results of measurement.

In recent years, the time required to make a noise frequency analysis in laboratory tests is being reduced to 1 or 2 minutes by the application of a panoramic harmonic analyser which automatically scans the noise spectrum on the screen of a cathode-ray oscillograph. One drawback of such analysers is that they are comparatively cumbersome.

In a more detailed study of the causes of noise it is advisable to consecutively determine the noise level for the various members of the kinematic chain or the units of the machine tool being investigated and to draw up a noise level balance. On the basis of this balance and a comparison of the design values of frequencies of the forced and free vibrations with the frequency analysis results, it is possible to reveal the causes of noise at the given speed step and to contemplate measures for their elimination. At the same time, the mean values of the noise level for various elements of the machine are determined; they are accepted as norms.

In cases when the required sound-analysis apparatus is not available, ENIMS recommends that the noise be recorded on magnetic tape which can be sent for further treatment to a laboratory or plant having the necessary equipment.

CHAPTER 31

INSTALLING METAL-CUTTING MACHINE TOOLS ON A FOUNDATION

The performance of a machine tool and its service life depend to a considerable degree upon its proper installation on the foundation. The foundation enables the load due to the weight of the machine tool and the mounted workpiece to be uniformly transmitted to a larger area of soil, and the required position to be quickly imparted to the machine tool in levelling operations. In the installation of most lathes, vertical turret lathes, drill presses and other machines of standard accuracy with predominant static loads, a weight up to 1.5 or 2 tons and a sufficiently rigid bed or column (if its length l to height h ratio does not exceed about 2), no other requirements are usually made to the foundation. As a rule, the foundation of such machine tools is the concrete floor of the shop with a thickness from 150 to 250 mm. If the shop has no concrete floor, separate concrete slabs, up to 300 mm thick and 4×4 m^2 in plan, can serve as foundations. If the weight of the installed machine is small and there are no or only insignificant dynamic loads during operation, the size of the foundation may be chosen on the basis of design considerations without resorting to tentative calculations. Quite frequently, particularly in installing special machine tools united in a production line, concrete strips 1.5 to 3 m wide and up to 6 m long or a continuous concrete strip 300 to 400 mm thick can be used as a foundation. The checking calculation for such foundations usually consists in determining the deflections f_i at various points for a given load on the basis of the theory of beams or slabs resting on an elastic base. Then the pressure on the soil is determined. Thus

$$\sigma = cf_1$$

where c is a coefficient depending upon the nature of the soil. The beds or bases of large and heavy machine tools are frequently insufficiently rigid. In such cases a supplementary requirement is made to the foundation; it is secured to the bed to form a common closed circuit thereby increasing the rigidity of the bed. Various types of foundation bolts are used to secure the bed.

Finally, in the installation of precision machine tools or machine tools in which dynamic loads predominate, the foundation has still another func-

tion—to protect the machine against external vibration and to reduce the frequency of natural vibrations of the system by increasing its total mass. This reduces the amplitude of forced vibrations of the machine. Most typical in such cases is an individual foundation in the form of a single monolithic slab from 0.6 to 0.9 m high.

Foundation calculations usually begin with the drawing of a design diagram and the determination of the magnitudes of the forces acting on the foundation and the co-ordinates of their points of application. As a rule, dynamic loads are reduced to equivalent static loads as follows:

1. In rotary motion, when the dynamic load is due to the centrifugal forces of unbalanced rotating masses of the machine, the equivalent static load can be determined approximately from the formula

$$P_{eq}=\frac{G}{g}\,\varepsilon\left(\frac{\pi n}{30}\right)^2 k_{dyn}$$

where G = weight of the rotating masses
g = acceleration of gravity
ε = eccentricity of the rotating masses (it can be assumed that $\varepsilon = 0.1$ of the workpiece diameter)
n = speed of workpiece, rpm
k_{dyn} = 1.5 to 2 = dynamic coefficient.

2. In reciprocating motion, the equivalent static load can be taken five or six times the value of the cutting force.

After drawing up the design diagram, the dimensions of the foundation are determined. On the basis of the installation or erection drawing, which shows the general overall dimensions and those required for installation, the distance between the pockets for the foundation bolts, etc., the outline and dimensions of the foundation are determined in the plan view. The outline is simplified to the greatest possible degree and, at the same time, all efforts are made to keep the centre of gravity of the machine tool and foundation from being displaced, in respect to the centre of the footing of the foundation, by more than 0.03 or 0.05 of the size of the side of the footing, in the direction of the displacement.

The weight of the foundation can be determined approximately from the empirical formula

$$G_f = k_f G_{mt}$$

where G_{mt} = weight of the machine tool
k_f = empirical factor, taken from 0.6 to 1.5 for machine tools with a static load and 2 to 3 for those with a dynamic load.

Knowing the weight G_f of the foundation, the area of its base F_f and the specific weight γ_f of the material, the height of the foundation can be cal-

culated. Thus

$$H_f = \frac{G_f}{F_f \gamma_f}$$

The height H_f of the foundation is to be checked in various ways.

1. To prevent the soil from being squeezed out at the sides of the foundation, its height H_f should satisfy a formula derived by S. Belzetsky

$$H_f \geqslant \frac{\sigma_z}{\gamma} \tan^4\left(45° - \frac{\varphi}{2}\right) - b\frac{1-\tan^4\left(45° - \frac{\varphi}{2}\right)}{2\tan\left(45° - \frac{\varphi}{2}\right)} \text{ m}$$

where σ_z = pressure on the soil, kgf/m², obtained under the footing of the foundation with the selected dimensions
γ = specific weight of the soil, kgf/m³
φ = angle of repose of the soil, deg
b = width of the foundation, m.

2. The height H_f should be sufficient to accommodate the foundation bolts whose length is determined from the condition of equal strength of the bolt in tension and of a part of the foundation being torn away in the form of an inverted truncated cone. Depending upon the material of the foundation, the length l of the foundation bolt should be (15 to 20) d_1, where d_1 is the minor diameter of the thread on the bolt.

3. The height H_f of the foundation of heavy machine tools should be greater than the freezing depth in the given locality.

4. To eliminate the action of foundations on each other due to soil settlement, the angle φ between the footings of adjacent foundations should be less than the angle of repose of the soil (Fig. 328*a*). The angle of repose is the acute angle which the line of the slope makes with the horizontal surface at the base of an embankment. This angle ranges from 15° to 20° for moist or rich clay and up to 50° for dry loam. The average value is $\varphi_{ar} \cong 40°$.

After determining and checking the geometrical dimensions of the monolithic foundation slab:

1. The average pressure σ_z of the foundation on the soil is compared with the permissible pressure αR:

$$\sigma_z = \frac{\sum P_z + G_f}{F_f} < \alpha R$$

where ΣP_z is the sum of all the vertical forces exerted on the foundation by the machine tool, with all the reduced dynamic loads taken into consideration. Coefficient α ranges from 0.8 to 1 for machine tools. Depending upon the soil, its porosity and saturation, R ranges from 1 to 6 kgf per sq cm.

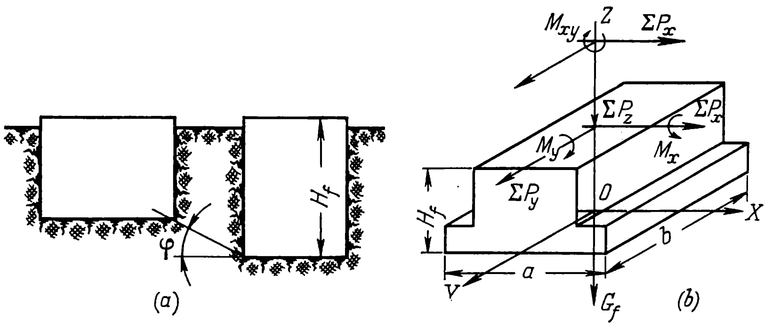


Fig. 328. Foundation design diagrams:
(a) dependence of the depth of a foundation on that of adjacent foundations; (b) stability of a foundation in respect to overturning

2. The stability of the foundation is checked, in respect to overturning about edges a and b (Fig. 328b), by calculating the corresponding stability factors:

$$k_{sb\ a} = \frac{M_{sb\ a}}{M_{ot\ a}} = \frac{\left(\sum P_z + G_f\right)\frac{b}{2}}{\left(M_x + \sum P_y H_f\right)}$$

$$k_{sb\ b} = \frac{M_{sb\ b}}{M_{ot\ b}} = \frac{\left(\sum P_z + G_f\right)\frac{a}{2}}{\left(M_y + \sum P_x H_f\right)}$$

where M_{sb} and M_{ot} are the stability and overturning moments, respectively. The foundation is considered stable if

$$k_{sb} \geqslant 1.8 \text{ to } 2$$

3. The foundation is checked for resonance. The frequency (cps) of the natural vibration of the foundation with the machine tool is determined from the following formulas:
for vertical vibrations

$$f_{z0} = 0.16\sqrt{\frac{k_z}{M}}$$

for horizontal vibrations

$$f_{x0} = 0.16\sqrt{\frac{k_x}{M}}$$

for rocking motion

$$f_{\varphi 0} = 0.16\sqrt{\frac{k_\varphi}{I_m}}$$

where k_z, k_x and k_φ = stiffness factors equal to

$$k_z = c_z F_f \text{ kgf per m}$$
$$k_x = c_x F_f \text{ kgf per m}$$
$$k_\varphi = c_\varphi I \text{ kgf-m}$$

c_z = coefficient of elastic uniform compression of the soil, kgf per cu m

$c_x = 0.5c_z$ = coefficient of elastic sheat of the soil, kgf per cu m

$c_\varphi = 2c_z$ = coefficient of elastic nonuniform compression of the soil, kgf per cu m

F_f = area of the foundation footing, sq m

M = mass of the foundation together with the machine tool, kgf-sec^2 per m

I = moment of inertia of the foundation footing in respect to an axis passing through the centre of gravity and perpendicular to the plane of vibration, m^4

I_m = moment of inertia of the mass of the foundation together with the machine tool in respect to the same axis, kgf-m-sec^2.

To exclude the danger of resonance, at which the amplitudes of vibration of the machine tool may increase to impermissible values, there should be a definite interval between the frequencies f_0 of the natural vibrations of the foundation with the machine tool and the frequencies f of the disturbing periodic forces.

Vibration isolation, i.e., attenuation of the vibration, occurs at ratios of $\frac{f}{f_0} > \sqrt{2}$. At $\frac{f}{f_0} < \sqrt{2}$, the vibration will build up; it reaches maximum intensity at $f = f_0$ (resonance).

The chief material used for making foundations for machine tools is concrete of grade 75 and stronger (usually 75, 90 and, for critical foundations, 110). The number designating the grade of concrete indicates the compression strength in kgf per sq cm after having set for 28 days. Concrete foundations for machine tools weighing over 10 or 12 tons, as well as those for machine tools subject to high dynamic loads, are to be reinforced with a steel network made of round bars laid under the footing of the bed at a distance of 20 to 30 mm. If the foundation is of great extent it can be reinforced with two networks (the second at the footing of the foundation). The concrete for a reinforced foundation should be of grade 100 or stronger. Sometimes, at a load exceeding 0.5 kgf per sq cm, the concrete floor of the shop is also reinforced with a network.

In the case of strip foundations, it is better to employ a tougher concrete which can withstand considerable deformation without failure. Other pos-

sible materials for foundations are: quarrystone—cheap local stone of any kind with a strength of at least 100 kgf per sq cm, and well-baked brick. Brick foundations are liable to be damaged by ground water. Consequently, if such a foundation is to be laid at a depth below the ground water level, it should be water-proofed (for example, by asphalting the surfaces).

The making of a foundation begins with the excavation of the pit. At the bottom of the pit a pad of rammed sand, broken stone, slag or gravel is laid to even up the foundation footing. The form for moulding the concrete foundation is laid along the pit walls of smooth boards, if possible, without slits or cracks. In accordance with the erection drawing, wooden plugs are provided at the places where the foundation bolts are to be located. It is best to make the plugs sectional to facilitate their removal after concreting. The reinforcement is secured in the pit if it is specified in the job instructions. Vibrations are used in laying the concrete. The concrete is put down in horizontal layers with no interruptions in the work so as to obtain a well-knit monolithic structure.

From 3 to 5 days after concreting, when the strength of the concrete reaches 50 per cent of the design value, the machine tool can be erected on the foundation. It is practically impossible to install the machine with sufficient accuracy directly on the foundation since the top surface usually varies from the horizontal by at least 0.25 or 0.5 mm per 1,000 mm. Therefore, the machine tool is levelled on the foundation by using: (1) separate pads or spacers from 0.3 to 3 mm thick and levelling wedges for light and medium-weight machine tools of standard accuracy or (2) levelling shoes and heavy cast-iron foundation plates for heavy and precision machine tools. After installing and levelling the machine to a spirit level, the clearance between the foundation and machine tool (minimum 50 to 70 mm) is grouted with concrete of plastic consistency or cement mortar. Such a grouting is the simplest way of joining the machine with the foundation and is extensively used in installing most medium-size machine tools on the concrete floor of a shop. Before grouting, a form of boards 20 to 30 mm higher than the grouting level is assembled around the perimeter of the foundation. Foundation bolts are used for holding down the machine more reliably. The bolts are inserted through the holes in the bed so that in installing the machine the bolt heads enter pockets provided for this purpose in the body of the foundation. During grouting, the pockets are also grouted with cement mortar. After the concrete sets and the bolts are drawn up tight, the machine is sufficiently rigidly secured to the foundation. If anchor bolts are to be used, anchor plates should be provided at the bottom of the bolt pockets. Foundation bolts are calculated in tension, taking the initial tightening into account. Thus

$$1.35P = \frac{\pi d_1^2}{4}[\sigma]$$

where d_1 = minor diameter of the bolt thread
1.35 = factor taking the initial tightening into account
$[\sigma]$ = permissible stress of the bolt material in tension.

If the machine tools are installed on concrete strips 200 to 300 mm thick, on upper floors or in other places where foundation bolts cannot be used, the main method of holding down machine tools (in addition to grouting) is the use of erecting beds. Such a bed is made by laying several rows of channels or other beams so that T-slots are formed.

It is not advisable to start operation on a machine tool erected on a foundation before the concrete has set to reach a strength of 70 per cent of the design value (about 7 days).

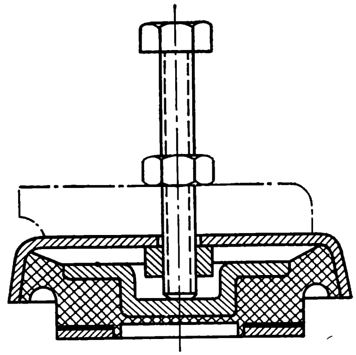


Fig. 329. Combined rubber and metal vibration isolator (Barry Controls Incorporated, USA)

Elastic vibration isolators and shock mounts are being widely used nowadays in the installation of machine tools that have a dynamic action on the surroundings, in the installation of high-precision machine tools susceptible to vibration of the foundation base, as well as many general-purpose models. Machine tools can be installed much quicker on vibration isolators, a better quality of surface finish is obtained on precision machines, and the noise and dustiness of the air in the shop are reduced. The use of isolators proves especially convenient in installing machine tools on upper floors, in rearranging machine tools to suit a changed manufacturing process, etc. These vibration isolators can be classified, according to the material of which the elastic element is made, as: (1) rubber (sheets and pads) and combined rubber and metal isolators; (2) all-metal isolators and mounts (with helical, flat or Belleville springs or knitted metal cushions); (3) felt pads (used, as a rule, for compressive loads and having a high damping effect); (4) cork pads (due to the comparatively high stiffness cork is chiefly used for sound proofing); (5) plastic pads (laminated, impregnated with vinyl plastics and allowing high adjustment); (6) pneumatic isolators.

An investigation of vibration isolators, conducted in ENIMS by E. Rivin, showed that combined rubber and metal isolators with the rubber in shear are the most efficient for the installation of machine tools. Shown in Fig. 329 by way of example is the rubber and metal isolator manufactured by the Barry Controls Incorporated (USA). The machine tool bed is mounted directly on the cover of the isolator. This cover is adjustable due to the deformation of the rubber rim upon turning the adjusting screw which bears on a metal thrust flange.

Depending upon the kind of rubber and the size of the isolator it can withstand a load from 60 to 2,700 kgf. The metal frame protects the rubber against the action of oils, solvents and sunlight, thereby prolonging the service life of the isolators. Rubber and metal isolators, in which the rubber is in compression, ensure vibration in the horizontal direction as well, isolation being even more efficient in the horizontal than in the vertical direction. This is due to the considerably greater flexibility when rubber is used under shear stress. Consequently, the natural vibrations of the machine on the isolators have a substantially lower frequency in the horizontal direction. This complies with the requirements made to the vibration isolation of machine tools.

Applying various types of elastic isolators and mounts, various natural frequencies of vertical vibrations f_0 of the machine tool can be obtained:

for $f_0 \geqslant 25$ cps—pads of rubber, cork, felt, plastics, etc.;

for 25 cps $\geqslant f_0 \geqslant 10$ cps—rubber and combined rubber and metal isolators with the rubber in compression;

for 10 cps $\geqslant f_0 \geqslant 5$ cps—rubber and metal isolators with the rubber in shear, and isolators with knitted metal cushions;

for $f_0 \leqslant 5$ cps—helical or flat steel springs.

The frequency f_0 of natural vibrations can be more exactly determined by the formula

$$f_0 = 5\sqrt{\frac{k}{\Delta_{st}}} \text{ cps}$$

where k = ratio of the stiffness of the isolator upon vibration to the stiffness in static loading (the so-called dynamic coefficient)

Δ_{st} = static deformation of the isolator from the weight of the machine tool, cm.

An incorrect choice of elastic isolators may lead to an intensification of the vibration if f_0 is close to the frequency f of the disturbing force. In the case of active vibration isolation (see next chapter), when the frequency f of the disturbing force can be readily determined, the frequency range $\frac{f}{f_0} > 2$ is used since it is sufficiently removed from resonance. This attenuates the vibrations three- or fourfold.

In the passive vibration isolation of precision machine tools, the value of f_0 is determined by special investigation (see next chapter).

CHAPTER 32

INVESTIGATING PASSIVE VIBRATION ISOLATION OF MACHINE TOOLS

One of the means of increasing the machining accuracy and improving the surface finish is to isolate the machine tool from the vibrations of the foundation base. This is called *passive* isolation as distinguished from *active* vibration isolation in which the external source of vibrations is isolated so that vibrations are not transmitted to the foundation. Passive vibration isolation is of prime importance, especially for precision machine tools. Employing seismic-type pickups, ENIMS conducted measurements on the vibration of shop floors in engineering plants due to the operation of metal-cutting equipment. These measurements showed that the amplitudes of vibrations of the base are comparatively large and may reach several microns or even hundredths of a millimetre. This leads to the appearance of waviness on the machined surface with a wave length determined by the frequency of the disturbance and a height approximately proportional to the amplitude of vibration of the base. Under such conditions, it is impossible to obtain workpieces of the required quality with tolerances equal to tenths of a micron unless passive vibration isolation of the precision machine tool is resorted to.

Rigid installation of a machine tool on its foundation—on wedges with subsequent grouting or on levelling shoes (see previous chapter)—does not always ensure the necessary vibration isolation.

A special isolated foundation, suspended on springs and having a natural frequency of vibrations of the order of 1.5 to 2 cps, achieves the desired effect but is too expensive and is therefore applied only for unique high-precision machine tools. For this reason, the promising and inexpensive method of installing precision machine tools on elastic vibration isolators has found such wide application in the last years. These devices are frequently used to install machine tools of standard accuracy as well (see Chap. 31).

Until recently, the phenomena occurring in machine tools upon vibration of the foundation base had not been investigated at all. The choice of one or another method of vibration isolation was entirely incidental, and the isolation could turn out to be either excessive or insufficient.

Research conducted in ENIMS, under the supervision of V. Kaminskaya, on the vibration isolation of machine tools threw some light on the question

of the susceptibility of certain types of machine tools to vibrations of the foundation base and established the effectiveness of a definite system of vibration isolation for given conditions.

In tentative investigations of the vibration of the foundation base of shops with equipment of various compositions, it was established on the basis of the results of statistic analysis that the spectrum of these vibrations is very broad and contains frequencies from 1.5-2 to 50-60 cps (since random disturbances turn out to be commensurable with periodic ones). Therefore, it is not always possible to achieve vibration isolation of a machine tool only by ensuring a large difference between the frequency of the external disturbances and that of the natural vibration of the machine system as a single mass (so-called resonance test). It is known that this (lower) natural frequency of the system depends upon the type of installation (rigid, on elastic isolators, etc.) which has no practical effect on the upper natural frequencies of the system. The relative vibrations of the cutting tool and workpiece, affecting the machining accuracy and surface finish, occur precisely at these higher frequencies which are determined by the natural frequencies of the units carrying the tool and workpiece. According to tentative data from ENIMS, the lowest of these upper frequencies range from 60 to 100 cps for lathes, from 30 (with antifriction ways) to 60 cps for surface grinders, etc. The vibration of the machine bed, usually at the lower natural frequency (except in the case of a rigid installation, when it repeats the vibration of the floor), is a kind of disturbance in respect to the units that carry the tool and workpiece. Any increase in the difference between the frequency of this disturbance and the frequencies of the units holding the cutting tool and workpiece leads to a considerable reduction in the amplitudes of their relative vibrations. The law of vibration transmission from the bed to the cutting zone characterizes, in essence, the quality of vibration isolation provided for a machine tool.

It has been proposed to assess the effectiveness of vibration isolation by the transfer factor, i.e., the extent to which vibrations are transmitted from the foundation base into the cutting zone at various frequencies, corresponding to the natural frequencies of the machine tool on the isolators or mounts. The transfer factor k is the ratio of the amplitudes of relative vibrations between the cutting tool and workpiece (a_{rel}) to the amplitudes of vibration of the foundation base (a), i.e., $k = \frac{a_{rel}}{a}$.

Assuming only a weak interlinkage between the vibrations of the machine tool on the isolators and the vibrations of its upper units, as is usually the case, the transmission factor can be expressed as the product

$$k = k_1\gamma = \frac{a_{bed}}{a}\frac{a_{rel}}{a_{bed}}$$

where k_1 = vibration transfer factor from the foundation base to the bed
γ = vibration transfer factor from the bed to the cutting zone
a_{bed} = amplitude of vibrations of the bed, measured at a definite place.

This procedure enabled the overall transfer factor to be broken down into the part determined chiefly by the parameters of the isolators or mounts and the machine tool as a single mass (factor k_1), and the part depending only upon parameters of the machine. The latter, factor γ, essentially characterizes susceptibility of the machine to vibrations of the foundation base. In the simplest case, when the vibrations of the units of the machine tool are being considered only in a plane perpendicular to the machined surface (these being the ones affecting the workpiece accuracy to the greatest extent), it becomes necessary to take into account the vertical (along the z-axis), horizontal (along the y-axis) and the rocking vibrations of the bed. Then the relative vibrations of the cutting tool and workpiece will be caused by all three kinds of disturbances. There is usually a weak interlinkage between the vertical and horizontal vibrations, and a strong interlinkage between the horizontal and rocking vibrations. This allows only two factors to be determined separately: γ_z—coefficient of sensitivity of the machine to vertical vibrations of the bed, and γ_y—coefficient of sensitivity to horizontal vibrations, taking into consideration their interlinkage with the rocking vibrations.

In this manner, the experimental investigations in the sensitivity of machine tools to vibrations of the foundation base consisted in the measurement of the relative vibrations of the cutting tool and workpiece (a_{rel}), absolute vibrations of units of the machine tool, mainly the bed (a_{bed}), and the vibrations of the floor in the vicinity of the machine tool (a). The last of these either appeared in the experiments from the operation of neighbouring equipment, or they were artificially produced by bumping the bed, dropping a heavy ingot near the machine, or using a centrifugal vibrator installed on the floor next to the machine.

A centrifugal vibrator (one designed by G. Levit was used) consists of a pair of engaged toothed gears with a ratio of $i = 1$ (Fig. 330). Holes are provided along the periphery of the gears into which a definite number of pins (weights) can be inserted in a definite order. The gears are assembled and the pins are arranged in the holes, symmetrically to a vertical plane passing through the point of contact of the pitch circles of the gears, in such a way that the horizontal components of the centrifugal forces of the pins compensate one another while the vertical components are added together. This vibrator, of comparatively simple design and sufficiently powerful, generates strictly vertical vibrational forces. The frequency of vibration is varied by changing the rotational speed of the gears. This can be done very simply since the vibrator is driven by a hydraulic axial-piston motor, model МГ-151, providing for variable-displacement speed control in a stable speed

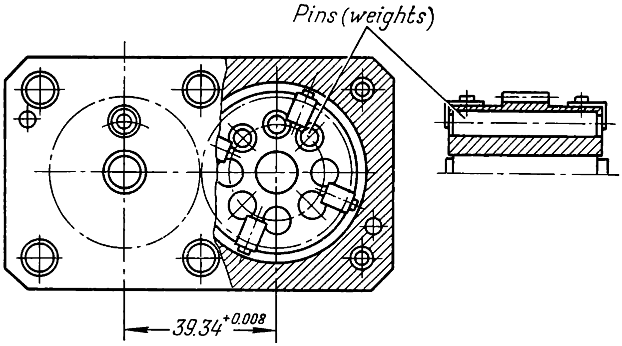


Fig. 330. Centrifugal vibrator (designed by G. Levit of ENIMS)

range from 40 to 3,600 rpm. The amplitude of force action is changed by changing the material of the pins (aluminium, steel or lead), their number and their arrangement in the holes of the gears. A drawback of centrifugal vibrators is the dependence of the amplitude of excited vibrations on the frequency, since the centrifugal force is equal to

$$F_c = m\omega^2 r$$

where ω is the angular velocity.

Lathes, surface grinders, cylindrical grinders and milling machines of various models were investigated. They were installed on wedges or shoes, grouted with cement mortar, on a spring-suspended foundation or on various types of elastic isolators. Investigations covered not only machine tools installed on the ground floor, but those on upper floors as well. To compare the results of the experiments, the positions of the units on machine tools of the same type were the same and corresponded to some typical case in machining practice.

In cutting, the machine tool is affected by a whole series of vibration exciters, and it becomes difficult to single out the influence of the vibrations of the foundation base. Assuming as a first approximation that the machine tool constitutes a linear system and operates in a range far from instability, it becomes possible to consider the action of various exciters separately, i.e., to conduct the experiments without cutting and with the machine not running. The validity of these assumptions is substantiated by the definite relationship existing between the amplitudes of relative vibrations of the tool and workpiece, measured when the machine tool is not in operation, and the waviness of surfaces machined in the same machine tool.

The vibration of the beds and foundation bases was measured by seismic-type pickups. New low-frequency transducers of the seismic type with a

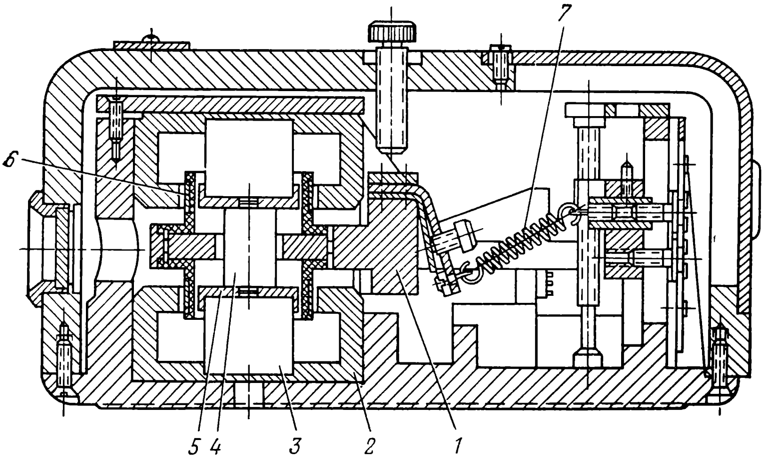


Fig. 331. Low-frequency induction transducer of the seismic type made by the Vibropribor Plant of Kishinev

galvanometric recorder have been developed in the USSR in recent years. One of these—model K-001 made by the Vibropribor Plant of Kishinev—provides an amplification factor of 1,000 in the range from 2 to 200 cps, thereby enabling vibrations with a minimum amplitude of 0.7 to 1 micron to be efficiently recorded.

A longitudinal section of such a transducer is shown in Fig. 331. The casing contains a fairly heavy pendulum *1* hanging from a special suspension. Two cylindrical coils *6* are secured on the free end of the pendulum. Each coil is located in its own magnetic system consisting of magnetic circuit *2* and permanent magnet *3* with pole shoes *5*.

The magnetic system is mounted in the casing. Brass cylinder *4* is arranged between the magnets to locate them more accurately and hold them more securely. Pendulum *1* is held in the equilibrium position by spring *7*.

When the transducer is placed on the vibrating item, it generates an electromotive force proportional to the velocity of the displacement. In order to record the vibrational displacements themselves, the voltage from the transducer is applied through a suitable controlled amplifier on a special integrating vibration-type moving-coil galvanometer (loop). A feature of these galvanometers is that they operate under well overdamped conditions. Therefore, upon the application to the galvanometer of a voltage proportional to the velocity of displacement, the deflection of the galvanometer will be proportional to the displacement itself. The galvanometer (loop)

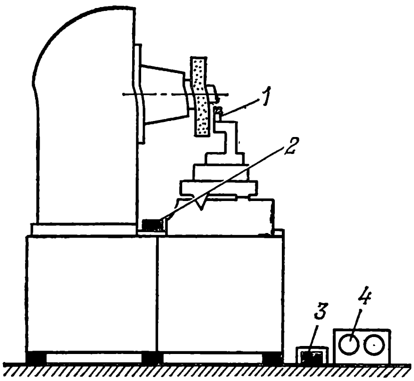


Fig. 332. Instrumentation setup for investigating the vibration isolation of a surface grinder:

1—head of the vibration transducer for measuring a_{rel}; 2—seismic-type transducer for measuring a_{bed}; 3—seismic-type transducer for measuring a; 4—centrifugal vibrator

is mounted in the casing of a loop oscillograph, type H700, and the displacements are recorded on light-sensitive paper.

The relative vibrations between the cutting tool and the workpiece were measured by the head of the type K-61A vibration transducer with wire resistance pickups (see Figs. 311 and 312).

A typical instrumentation setup for the vibration isolation investigation of a surface grinder is shown in Fig. 332.

After suitable treatment, the measurement data were used to plot gain-frequency characteristics (GFC) of the elements of the machine tool (Fig. 333) for various methods of installation. Of greatest interest for assessing the effectiveness of the applied system of vibration isolation and the sensitivity of the machine to vibrations of the foundation base were the sections of the GFC in the vicinity of resonances of some kind of vibration of the bed. The GFC and the oscillograms of free vibrations of the elements being investigated in the machine tool enabled curves to be plotted showing the dependence of the transfer factors k_1 and γ upon the frequency.

The curve γ vs f is shown in Fig. 334 for a surface grinder. The factor γ_z is increased at the frequency $\simeq 30$ cps because this is approximately the natural frequency of horizontal vibrations of the saddle on the antifriction ways (the frequency is determined by the rigidity of the feed mechanism).

The values of factor k_1 for vertical vibrations, obtained in the experiments, varied in the range from 5 to 10, the lower value of k_1 corresponding to a higher damping capacity of the isolators or mounts.

The overall vibration transfer factor k was about 0.1 when the machine tool was installed on elastic isolators with a natural frequency of 15 cps, and about 1.0 in case of a rigid installation.

The plotted curves and derived relationships enabled the principal parameters of systems of vibration isolation to be selected for high-precision machine tools. This was done in the following manner. It is known that the amplitudes of vertical vibrations of the floor in shops where precision machine tools are installed rarely exceed 1.5 or 2 microns. We shall assume tentatively that to obtain a surface finish of the 10th class (according to the USSR Std) with an average height of microirregularities $R_a = 0.16$ micron,

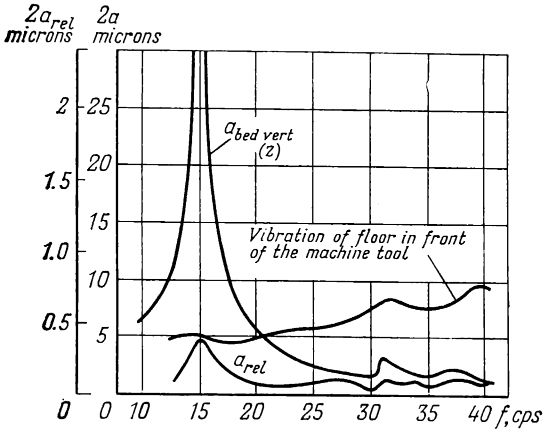


Fig. 333. Gain-frequency characteristics (GFC) of a model 3Б71M surface grinder installed on elastic isolators

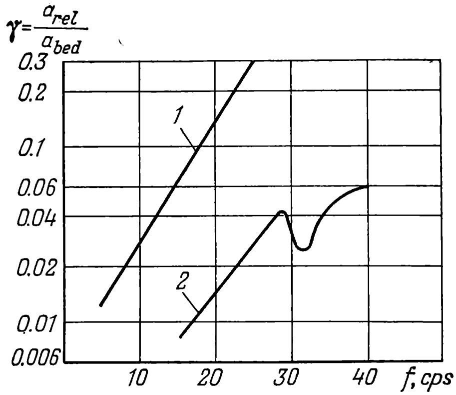


Fig. 334. Experimental values of the factors for the transmission of vibrations of the bed to the cutting zone of a model 3Б71M surface grinder:

1 and 2—transfer factors for the horizontal (γ_y) and vertical (γ_z) vibrations of the bed

the amplitudes a_{rel} should not exceed 0.5 R_a, i.e., 0.08 micron. Then the transfer factor at which the 10th class of surface finish can be obtained is

$$k = \frac{a_{rel}}{a} = \frac{0.08}{(1.5 \text{ to } 2)} \cong 0.05$$

It being known that $k_1 = 5$ to 10, we can determine the transmission factor for vertical vibrations from the bed to the cutting zone. Thus

$$\gamma_z = \frac{k}{k_1} = \frac{0.05}{(5 \text{ to } 10)}$$

Assuming a definite case in which $k_1 = 5$, we can find from the curve in Fig. 334 that $\gamma_z = \frac{0.05}{5} = 0.01$ is valid for a frequency of vertical vibrations of the bed on the isolators of the order of 17 cps. Knowing this frequency it is possible to make recommendations on the choice of a method for installing the machine tool.

The accumulation of data on the values of transfer factors γ at various frequencies for various types of machine tools makes it possible to choose a system of passive vibration isolation for high-precision machine tools more correctly and with greater accuracy.

CHAPTER 33

POWER TESTS FOR MACHINE TOOLS

The merits of a machine tool drive, as to its design, manufacture, assembly and production capacity, are characterized, in addition to other factors, by the efficiency of the machine.

Power tests are conducted to determine the efficiency as well as the friction losses in the drive (idle-run power consumption N_{ir}). More reliable results can be obtained in the tests if they are preceded by certain preparatory measures such as: checking and adjusting all the main elements of the drive; checking the grade and amount of applied lubricant in accordance with the lubricating instructions; heating up the machine, for which purpose it is operated at the medium spindle speed (medium number of ram, table or slide strokes per minute), etc. Tests may be begun as soon as it is established that the idle-run power consumption has not changed for as least 10 or 15 minutes.

The power balance equation for a machine tool performing useful work can be written as

$$N_c = N_{li} + N_{ir} + N_{ll} + N_{ef} \text{ kW}$$

where N_c = power consumption of the electric motor
N_{li} = power loss in the electric motor
N_{ir} = power loss in the machine tool when running idle
N_{ll} = power loss in the machine tool when under load
N_{ef} = effective (useful) power used in the cutting process.

Each of the values included in the balance equation varies with the speeds, feeds and other operating conditions. Power tests, in essence, amount to consecutive experimental determination of these values at varying operating conditions of the machine.

In the case of induction motors, their power consumption N_c is commonly determined by one of two methods: (1) two-wattmeter method (Fig. 335*a*) and (2) circuit with an artificial neutral point and one wattmeter (Fig. 335*b*).

The power consumption of a d-c motor is $N_c = UJ$, where U and J are the readings of the voltmeter and ammeter, respectively.

In most cases, the certificate is used to determine the power loss N_{li} in the electric motor. It indicates the motor efficiency values η_{em} at the rated

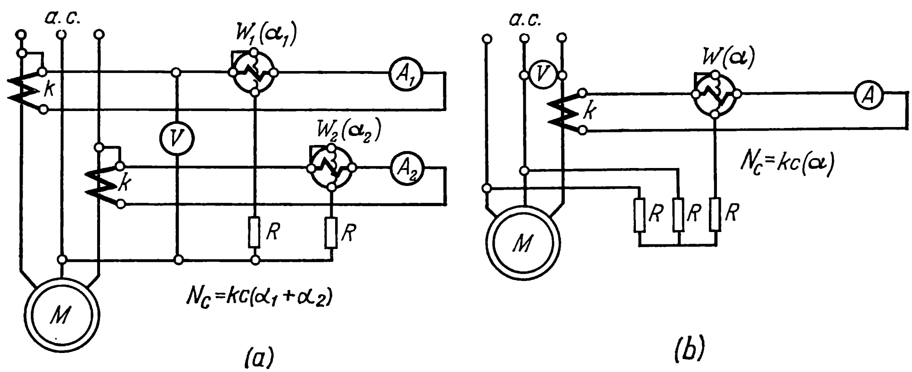


Fig. 335. Circuits for measuring the power N_c consumed by an electric motor from the power supply:

(*a*) two-wattmeter method; (*b*) circuit with an artificial neutral point and a single wattmeter; *k*—transformation ratio of the current transformers; *c*—factor depending upon the sizes of the series resistors *R*; *a*—wattmeter readings

power N_r as well as at 0.25 N_r, 0.5 N_r, 0.75 N_r, and 1.25 N_r. Using these data, no difficulty is encountered in plotting a loss curve, determining N_{li} from the equation

$$N_{li}=\frac{N_i}{\eta_{emi}}-N_i$$

where N_i = power developed by the electric motor

η_{emi} = efficiency of the electric motor at this power.

The value of N_{l0} (at zero load) can be determined by starting the electric motor for an idle run and measuring N_c by one of the above-mentioned methods. In this case, N_c is equal to the loss value sought for.

Sometimes the loss curve (Fig. 336) is given in the motor certificate. If no data whatsoever are available, the electric motor is demounted from the machine tool and its power output is measured by a braking method at loads of 0.25 N_r, 0.5 N_r, 0.75 N_r, 1.0 N_r and 1.25 N_r, determining N_{li} from the power

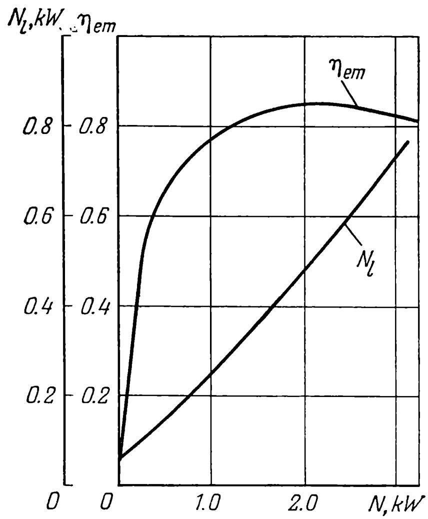


Fig. 336. Dependence of the efficiency η_{em} and power loss N_l in an electric motor on the power N developed on the motor shaft

balance equation for this case. Thus

$$N_{li} = N_c - N_b = N_c - \frac{M_b n}{975} \text{ kW}$$

where N_b = brake power
M_b = brake torque, kgf-m
n = speed of the electric motor measured with a tachometer, rpm.

The idle-run power consumption N_{ir}, which is directly concerned with the mechanical part of the machine tool, is determined with various degrees of accuracy and in more or less detail depending upon the kind of test being conducted.

In acceptance tests of lot-produced machine tools or of pilot models, N_{ir} is usually determined for all the speed steps from the power balance equation for idle running:

$$N_{ir} = N_c - N_{li}$$

where N_c and N_{li} are determined by one of the above-mentioned methods.

It is expedient to determine the idle-run power consumption of machine tools in laboratory tests at various conditions of lubrication and also for the various elements of the drive. The influence of the grade of oil and other factors on the losses in the gearbox can be cleared up by measuring N_{ir} as accurately as possible. Under laboratory conditions this is done in most cases using a torque motor (Fig. 337) with a rocking stator. When such a motor is used to drive the machine tool, a reactionary torque M_{st} is developed

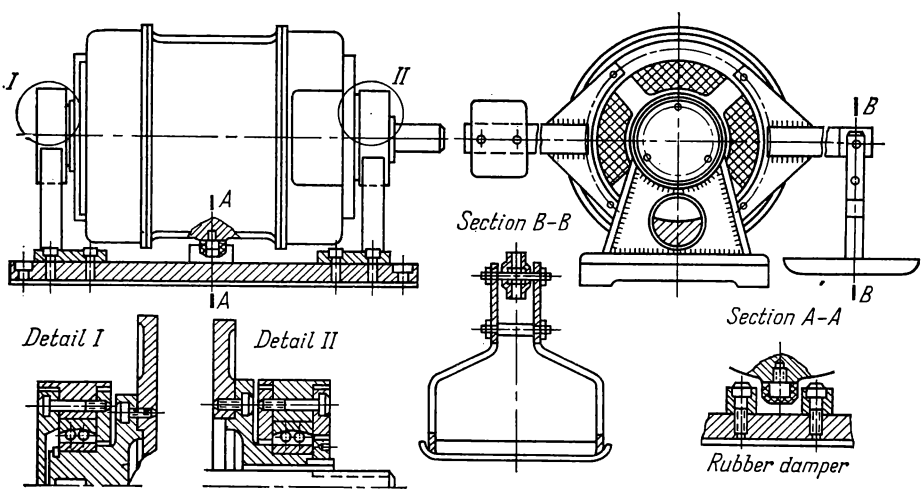


Fig. 337. Torque motor (electric dynamometer)

on its stator. This is counterbalanced by a weight, and the following relationship holds true:

$$M_{st} = M + M_0 \text{ or } M = M_{st} - M_0$$

where M = external torque on the electric motor shaft
M_0 = torque due to the resistance of the air and losses in the bearings upon idle rotation of the rotor in the motor.

Consequently, the power at the shaft of the torque motor, i.e., the power delivered to the machine tool from the motor, is

$$N = \frac{Mn}{975 \times 10^3} = \frac{(M_{st} - M_0)\, n}{975 \times 10^3} = \frac{(Q_{st} - Q_0)\, ln}{975 \times 10^3} \text{ kW}$$

where l = length of the scale arm, mm
n = speed of the motor, rpm; for greater accuracy it is necessary to have a curve of the relationship $n = f(Q)$
Q_{st} and Q_0 = counterbalancing weights, kgf, required in operation under load and in an idle run, respectively.

This formula can be simplified by making the length of the scale arm equal to 975 mm.

If the torque motor has been precisely manufactured, with the account of the error in rotational speed measurement, the power can be measured with an error within 1 or 1.5 per cent or even less.

The results of the measurement of the idle-run power consumption N_{ir} are used to plot the function $N_{ir} = f(n)$ to obtain the curves or family of curves which are the test data (Fig. 338). An analysis of the plotted curves enables the friction losses to be evaluated, the most favourable lubricating conditions to be established or the best version of drive design to be chosen.

To find which parts of the drive are most unfavourable in respect to friction losses, it is expedient to measure N_{ir} for separate sections and even for separate most critical elements of the kinematic train, excluding the remaining elements in some way. For example, the running-down method is used to investigate the friction losses in the spindle unit. Spinning the spindle, after disconnecting it from the rest of the drive, its braking process—running down—is recorded by means of any simple recording instrument. The running-down curve is then plotted from the measurement data (Fig. 339*a*).

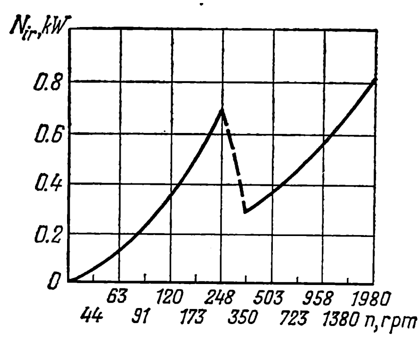


Fig. 338. Curve of the function $N_{ir} = f(n)$ for the model 1616 engine lathe

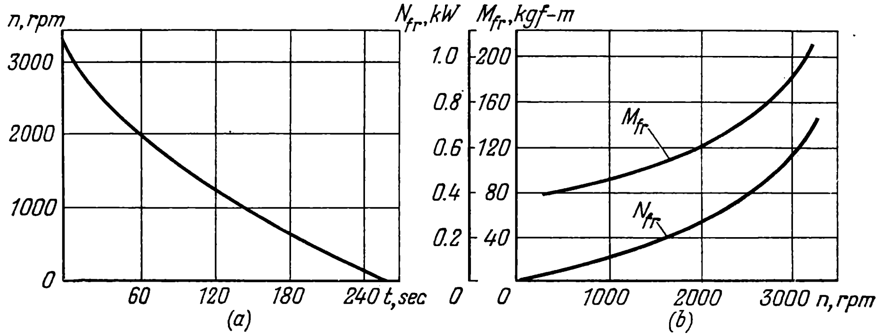


Fig. 339. Typical running-down curve (*a*) and curves showing the dependence of the friction power loss N_{fr} and the friction torque M_{fr} on the spindle speed n, rpm (*b*)

The motion equation for the idly running spindle can be written as

$$M_{in}+M_{fr}=0 \text{ or } \frac{\pi}{30} I \frac{dn}{dt}+M_{fr}=0$$

where M_{in} and M_{fr} = inertia and friction torques, respectively, kgf-m
I = moment of inertia of the rotating mass, kgf-m per sec²
n = speed of the spindle, rpm.

It follows that

$$M_{fr}=-\frac{\pi}{30} I \frac{dn}{dt}$$

By graphically differentiating the running-down curve (i.e., by finding $\frac{\Delta n}{\Delta t}$ at various moments of time) and determining I either experimentally or by calculation, it is possible to calculate the friction torque in kgf-m. Then, after calculating the friction power loss by means of formula $N_{fr}=\frac{M_{fr}n}{975}$ kW and plotting the curve $N_{fr}=f(n)$, the friction losses in the spindle unit will be clearly represented (Fig. 339*b*).

The next step in a machine tool power test is to determine the efficiency. Distinction should be made between the efficiency of the whole machine tool $\eta=\frac{N_{ef}}{N_c}$ and the efficiency of the transmissions of the machine tool drive (i.e., its mechanical part)

$$\eta_i=\frac{N_{ef}}{N_c-N_{li}}$$

where $(N_c - N_{li})$ is the power developed on the shaft of the drive motor.

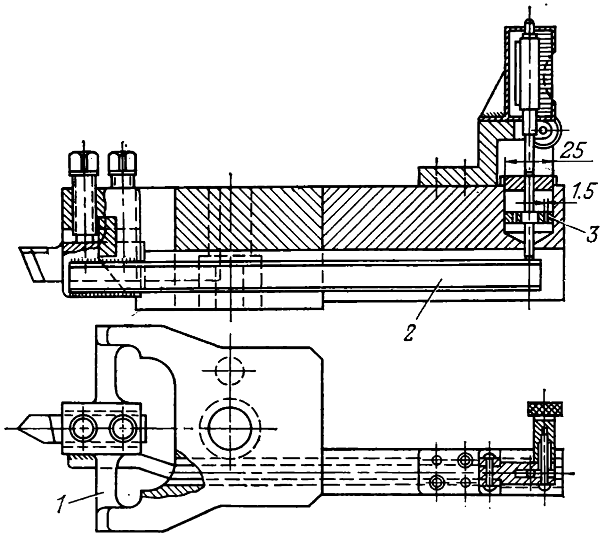


Fig. 340. Single-component dynamometer for measuring the cutting force designed by G. Levit of ENIMS

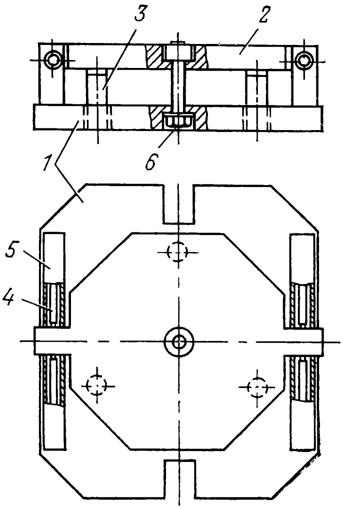


Fig. 341. Principle of a two-component drilling dynamometric table designed by V. Vasilyev

The effective (useful) power N_{ef} can be measured either in the cutting process or in braking. The former of these is more correct since the conditions of loading differ in braking and in operation. In machine tool tests it is desirable to use the cutting method. The effective power or the power required in cutting is determined from the equation

$$N_{ef} = \frac{Pv}{6{,}120} \text{ kW}$$

where P = component of the cutting force in the direction of the cutting motion, kgf

v = cutting speed, m per min.

In machine tools with a rotary primary cutting motion, it is sufficient to know the average value of the cutting force as determined by a mechanical, hydraulic or electrical single-component inertia dynamometers. One such dynamometer, designed by G. Levit of ENIMS, is illustrated in Fig. 340. In this instrument, the cutting force, acting through the cutting tool, twists the torsion bars *1*. This tilts bar *2*, and the previously calibrated dial indicator, whose spindle bears against the end of bar *2*, shows the average value of the cutting force. The provision of the simple damping device *3* smooths out the oscillations of the indicator.

It is convenient to measure the drilling torque in a drill press by means of a dynamometric table mounted on the press table.

The principle of a dynamometric table, designed by V. Vasilyev for this purpose, is illustrated in Fig. 341. In drilling, the torque is applied to upper plate *2* of the table and is withstood by elastic elements *4* which are secured on uprights *5* of body *1*. The elastic elements are hollow rods on which wire pickups are glued. These pickups are connected together differentially, i.e., so that the torque causes equal, but unlike (in the sense of positive and negative), deformation of each of the two groups of pickups of the measuring bridge. Each elastic element is secured on thread in body *1* of the instrument. This enables the lugs of plate *2*, located between the hollow bars, to be tightly clamped. The high preload thus produced raises the rigidity of the transmitting system of the dynamometer. Three elastic elements *3* withstand the axial force. They are preloaded by a special device *6*.

In milling machines, the dynamometer is usually built into the body of the milling cutter.

The effective power N_{ef} of machine tools with a reciprocating primary cutting motion (planers, shapers and slotters) is determined by measuring the instantaneous cutting force. As a rule, therefore, the cutting force is measured in this case by a special dynamometer having variable-reluctance or capacitive pickups. Wire-type pickups, glued to some elastic element of the dynamometer, can also be conveniently employed. The pickups are connected to amplifiers from which the signal is transmitted either to an oscillograph or to a very sensitive indicating instrument.

When N_{ef} is measured by the braking method, the torque on the spindle is developed and measured by a brake. The simplest of these is the Prony brake which may be a combined shoe and band brake as shown in Fig. 342. Hollow pulley *6* is mounted on the machine tool spindle. In the braking process, when a large amount of heat is evolved, cold water is delivered continuously through a tube into the inside of the pulley. The water is discharged through another tube whose end is turned to oppose the direction of pulley rotation. The dimensions of the pulley can be tentatively determined from the empirical formula

$$Db = (25 \text{ to } 50)\, N$$

where D and b = diameter and width of the pulley, respectively, cm
N = brake power, kW.

The lower values of this coefficient are taken for high speeds and low pressure on the brake band.

Prior to the test the brake is carefully balanced by inserting a triangular prism between pulley *6* and shoe *5*, and adjusting weight *1* along its beam. Then the hook at the right end of arm *2* is connected to a dynamometer

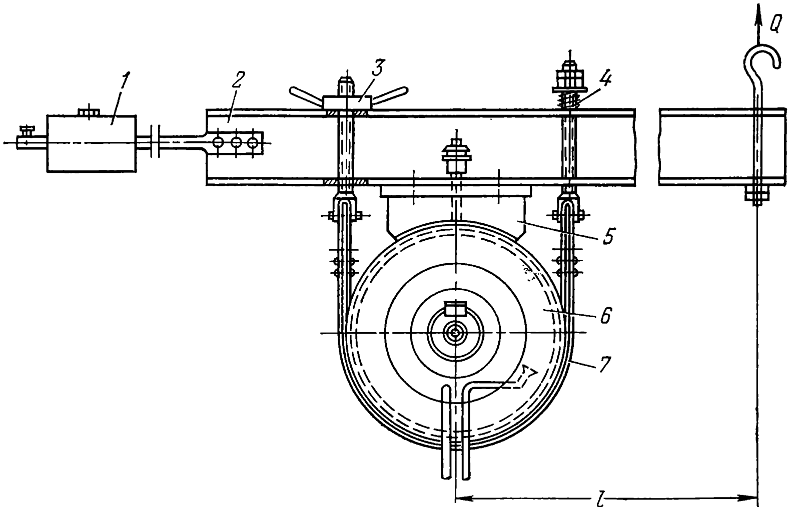


Fig. 342. Combined shoe and band Prony brake

for measuring the braking force Q. More efficient braking is obtained by lining shoe *5* and band *7* with a suitable friction material.

The brake is loaded by tensioning band *7* when nut *3* is tightened on its screw. Spring *4* provides for smoother loading.

The braking torque is determined by the formula

$$M = \frac{Ql}{1,000}\ \text{kgf-m}$$

and the brake power is

$$N = \frac{Qln}{975 \times 10^3}\ \text{kW}$$

If the arm l is taken equal to 975 mm, then

$$N = \frac{Qn}{1,000}\ \text{kW}$$

A brake of the design described here is very simple, but under certain conditions vibrations are set up when the surface of the pulley slides along the band. As a result, the braking torque cannot be maintained constant and the measuring accuracy is lowered.

In respect to smooth operation, electric brakes of the torque motor type are much more efficient. Most frequently d-c machines with separate excitations or induction motors with a phase-wound rotor are used as brakes.

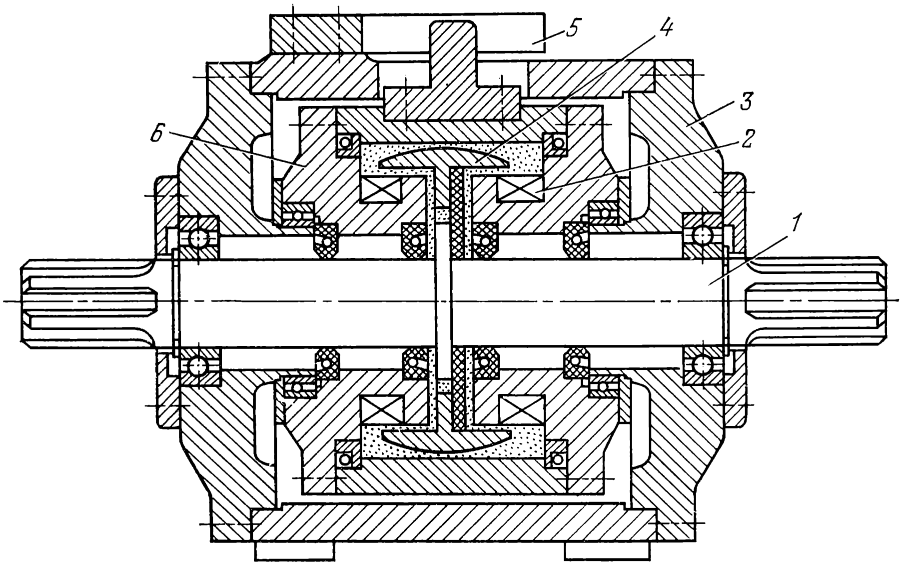


Fig. 343. Principle of a magnetic-particle brake developed in ENIMS

In the first case the braking moment is regulated by a rheostat connected into the field or armature circuit, and in the second by a rheostat in the rotor circuit.

Recently, a size range of magnetic-particle brakes, type ПТ, has been developed by ENIMS. They have been specially designed for testing machines. The principle of such a brake is illustrated in Fig. 343.

A low-inertia rotor *4* is secured rigidly on central shaft *1* of the brake. It is separated from stator *6* by a working gap of 1 to 2 mm. This whole gap is filled with an oil slurry of ferromagnetic particles (carbonyl iron powder mixed with oil). Rubber seals are provided on the shaft and in the stator to prevent leakage of the slurry. When magnetizing coils *2*, coaxial with the central shaft, are energized (the coils are connected in series and are supplied with direct current), the magnetic flux induced in the magnetic circuit—rotor and stator—cuts the gap. Electromagnetic forces acting in the gap increase the viscosity of the ferromagnetic slurry to a greater or lesser extent (depending upon the ampere-turns of the magnetizing coils). This interlinks the stator and rotor of the brake. Through shaft *1* and a spline coupling, rotor *4* of the brake can be linked to a mandrel fastened in the spindle of the machine tool. Stator *6* can rotate freely in housing *3* about the central shaft since it (the stator) runs on ball bearings. The reactionary torque on the

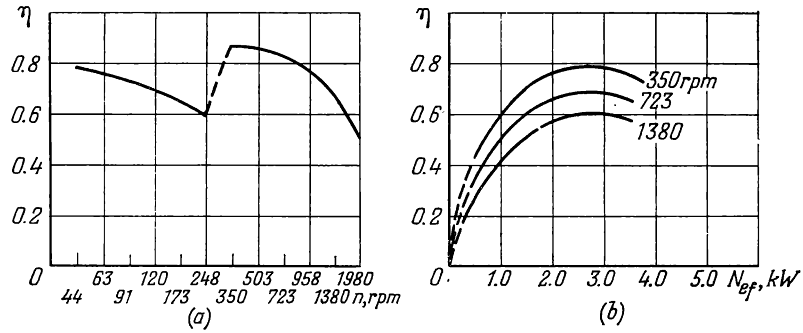


Fig. 344. Efficiency curves of the model 1616 engine lathe:
(*a*) at various speed steps (input power 4.3 kW); (*b*) depending upon the effective power N_{ef} developed on the spindle

stator, equal in magnitude to the braking torque developed on the spindle, is transmitted to dynamometer *5*, mounted on the brake housing.

A magnetic-particle brake of this type has a high speed of response, enables the torque to be varied according to any given law and ensures a constant instantaneous value of the torque. The brake is water-cooled.

The efficiency of a machine tool does not remain constant if the operating conditions are changed. Hence, in these tests it is advisable to measure the efficiency at least at three speed steps in the lower, medium and upper parts of the speed range. At each speed step, the efficiency is usually measured at several values of the power developed on the spindle corresponding to that on the electric motor shaft, equal to 0.25, 0.5, 0.75, 1.0 and 1.25 of the rated power. In making measurements at the lower speed step, when it may be impossible to use the full available power, the efficiency is measured at the maximum permissible torque. The results of these measurements are plotted as the curves $\eta = f_1\ (n)$ and $\eta = f_2\ (N_{ef})$ as shown in Fig. 344*a* and *b*, respectively.

As a rule, the efficiency increases with N_{ef} at each definite speed step, the rate of this increase being reduced as N_{ef} grows. At a constant power, the efficiency drops, as a rule, with an increase in speed; at low speeds, an increase in efficiency is sometimes observed.

An analysis of the plotted curves enables a conclusion to be drawn on the processing capacities of the machine tool. It enables the design and quality of manufacture and assembly to be assessed, and effective measures to be taken to reduce friction losses.

CHAPTER 34
SERVICE TESTS AND INVESTIGATIONS OF MACHINE TOOLS

It is an important and complex task to ensure trouble-free operation of a modern machine tool over a long period of service. Any trouble that interrupts normal operation even for a short time may lead to material losses, not to mention the expenditure for premature repairs. The problem of ensuring dependability and durability of production equipment in general, and of metal-cutting machine tools in particular, has become one of prime importance at the present time.

The conceptions of dependability and durability are closely related and together they characterize the working capacity or performance of a machine tool.

The principal criteria of machine tool performance are the static and fatigue strengths, wear resistance, capacity to withstand thermal deformation upon variations in the temperature, rigidity and vibration-proof properties.

A machine tool can be appraised in respect to the criteria of rigidity, vibration-proof properties, heat resistance and, in part, static strength on the basis of tests of comparatively short duration. On the contrary, prolonged service tests are required to assess the dependability and durability in respect to wear resistance, fatigue strength and static strength in connection with dynamic overloads. Up to the present time, an integrated appraisal of the dependability and durability of machine tools is not being made. Since 1963 a number of Soviet plants are conducting investigations in the performance of machine tools that are manufactured in large lots. Accordingly, a group of engineers of ENIMS, under the supervision of A. Lapidus, worked out a procedure for making such observations. This procedure enables the main defects deteriorating the durability and dependability of machine tools to be revealed, the causes of failures and damages to be established, the rate of wear of the principal parts to be determined and impairment of machine tool accuracy in the course of service to be noted.

Measures for ensuring the dependability and durability can be based on the results of such observations.

Service observations (tests) are carried out by a special group from the plant machine tool laboratory. The observations are made in a number of large plants in which machine tools of the models being investigated have

been installed and are in regular operation. Machine tools that are operating under the more typical conditions for the given model are chosen for observation. This may prove difficult since the working conditions of general-purpose machine tools (as to kind of jobs done, magnitude and repeatability of the loads, material being machined, configuration and rigidity of the blank, etc.) may vary in extremely wide ranges.

All data on the use of the machine tool during the observation period are noted in a special service observations logbook and are taken into consideration in the treatment of the results. More objective conclusions can be arrived at if observations are made on as many as 150 or 200 machines of the given model. To save time, tests are conducted, as a rule, on machine tools that operate in two shifts.

The conclusions based on the test results can be extended to cover all machine tools of the given model being manufactured only if the service tests are begun after the manufacturing plant has sufficiently refined the manufacturing process and assembly of these machine tools, i.e., when their quality has become sufficiently stable.

In accordance with a schedule, the staff of the test group make observations after each 3 or 4 months of regular operation of the machine tools, noting the data in the logbook. In the intervals between the observations made by the group, an account of all troubles (knocks, noises, vibration, overheating, etc.) and all downtime in the operation of the machine tool is to be kept by its operator who has received special instruction on carrying out observations.

When the whole cycle of observations has been completed, the obtained data are systematized and processed. The length of time the machine tool has been in operation is determined:

$$T_o = T_n - T_{dt} - T_{pm}$$

where T_n = nominal time budget during the period under investigation
T_{dt} = downtime of the machine tool due to repairs and adjustment
T_{pm} = downtime of the machine tool due to factors concerned with production management.

In the process of the service tests, an account is kept of all failures in operation which lower the level of performance of the machine. Failures in performance may be due to defects in the construction or manufacture of the machine or damages occurring in the course of operation. Defects and damages may be either systematic or occasional. Systematic defects and damages are of greater practical interest since they reflect features of the construction, quality of manufacture and assembly, and the service conditions.

A study of damages usually begins by establishing the circumstances under which they appeared. This is done by questioning the personnel; establish-

ing the periods (duration) and conditions (speeds and feeds) of operation, and carefully examining the damaged parts, especially fractured surfaces. Such an examination enables the kind of failure to be determined, data on the loading conditions to be obtained and, frequently, the stress raisers to be revealed that reduce the strength of the parts.

The main kinds of damage observed in machine tool components are wear, fatigue and static fractures, and permanent set. As a rule, wear predominates over all other kinds of damages. This ensues from the specific conditions of machine tool operation: a large number of movable joints or associations that frequently operate under conditions of boundary or even dry friction; operation under conditions in which a large amount of chips or abrasive dust is formed, etc. Therefore, especial attention is paid to the study of the wear of critical components in service observations.

The main types of wear are abrasion and seizing. The latter usually occurs at high sliding velocities and high pressures, when the temperature of the rubbing surfaces is high, and at low speeds with insufficient lubrication.

In most cases, the so-called linear wear is determined. This is the reduction, normal to the rubbing surface, in the linear dimensions of a part. Methods have been developed for expressing the amount of wear in service investigations of machine tools. Depending upon the purpose and concrete working conditions of the machine tool, the wear can be expressed by:

(1) The average rate of wear (for general-purpose machine tools performing most kinds of operations) as

$$i_v = \frac{W}{T_o}$$

where W = linear wear

T_o = period of operation of the machine tool (in months or years of two-shift operation).

(2) The referred average rate of wear (for general-purpose machine tools performing comparatively rare jobs, such as cutting threads, etc.) as

$$i_v = 100\frac{W}{\eta T_o}$$

where η is the time spent in performing the given operation expressed as a percentage of the total period of operation of the machine tool.

(3) The average intensity of wear (for special machine tools that machine a single definite workpiece) as

$$i_s = \frac{W}{S}$$

where S is the path of friction.

It is best to measure the wear without disassembling the machine and removing the parts. If this is impossible, measurements should be made during the next routine repairs.

The main methods employed to determine the amount of wear in machine tool tests are micrometry and the determination of local linear wear.

The *micrometry method* is the simpler of the two and is based on the measurement either of the change in diameter or other linear dimension of the part between two surfaces subject to wear, or the change in the distance from a constant datum to the surface being investigated. Any accurate surface, not subject to wear, can serve as the datum. In the first case, the amount of wear is measured by micrometer calipers, dial indicators, comparators, instruments with variable-reluctance and wire pickups of the feeler type or contactless pneumatic gauges for measuring linear dimensions. Special fixtures (of the stand type) with dial indicators find application in the second case.

The accuracy of wear measurements made by the micrometry method may be adversely affected by elastic and thermal deformation of the parts, as well as changes in the points of measurement in successive experiments.

In the method in which the local linear wear is determined, a converging indentation (depression) of definite shape is first made on the rubbing surface. Such indentations (50 to 75 microns deep and 1.7 to 2 mm long), spaced from 100 to 200 mm apart, have practically no effect on the operation of the rubbing surfaces. Changes in the dimensions of the indentations, taking place in the course of wear of the surface, are indications of the amount of linear wear at the given point.

The indentation (depression) on the friction surface can be made either by penetration with a square-base-pyramid diamond (indentation) method, or by cutting out a depression with a rotating diamond tool (cut depression method) as shown in Fig. 345. The second method is better because there is no bulging of the material at the sides of the indentation and little recovery in the layers of metal deformed in cutting. This allows more accurate measurements to be made. A special instrument has been developed by the Institute of Mechanical Engineering for measuring the wear of slideways by cutting out depressions. Its main parts are the attachment with the diamond tool and the optical system. The latter enables the length of the depression to be precisely measured before and after an established period of operation. The amount of local linear wear is determined as the difference in depth h of the depression, before and after wear, calculated by the formula

$$h = \frac{l^2}{8r}$$

where l = length of the depression

r = radius of curvature of the depression.

The period of observation for wear measurements usually constitutes from 6 to 18 months of operation in two shifts.

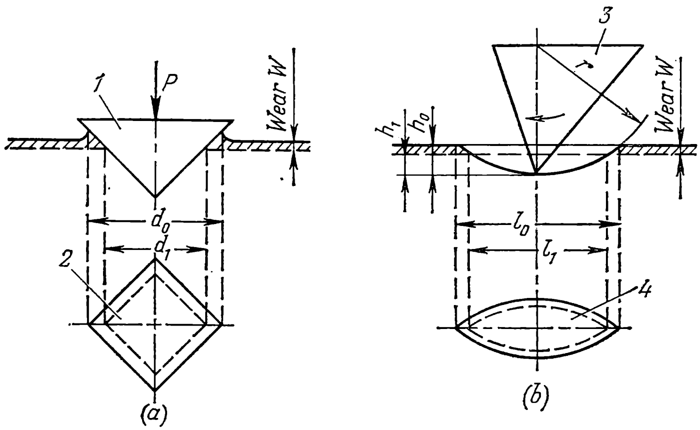


Fig. 345. Methods of determining the local linear wear W:
(*a*) indentation method; (*b*) cut depression method; *1*—square-base-pyramid diamond; *2*—indentation; *3*—cutting tool (triangular-base diamond); *4*—depression

After analysing the actual working conditions of the mating friction surfaces to establish the reasons for excessive wear (if such is observed), measures are planned to reduce the wear.

Another type of damage in machine tool parts—fatigue and static fractures—is more seldom encountered, since the dimensions of the parts are frequently determined, not by strength requirements, but on the basis of other performance criteria. Nevertheless, in connection with the measures being taken to reduce wear by improvements in design, as well as the increased speeds used in modern machine tools, higher dynamic loads and the reduction in the excess safety margin, the importance of such fractures has been steadily increasing.

Signs of fatigue failure are the characteristic zones on the fracture: seat of failure with one or several foci of fracture, zone of fatigue development and the failure finishing zone, and the definite service life of the part up to failure. Definite rules, in particular the structural scheme of fatigue fractures (according to I. Oding), enable the type of loading, magnitude of the load, and the size and kind of stress raiser to be established by examining the fracture.

After determining the kind of failure and the circumstances connected with the breakage, the causes of the failure are to be revealed. It has been proposed in ENIMS that all causes be arbitrarily divided into three groups which can be called design, manufacture and operation. It is still impossible to give any general recommendations enabling the true causes of failures to be

reliably determined. There may be different cause for the same kind of damage on the same kind of part. For this reason, investigations conducted in ENIMS list a series of typical examples of damages and indicate their causes.

Some of these examples follow:

"*Designer's errors*" *group*

Incorrect assignment of tolerances leads to jamming and breakage of the gears in the spindle drives of lathes due to the lack of sufficient backlash.

The gear breakage and twisting of the shafts in the speed gearboxes of various machine tools upon braking at the upper spindle speeds were due to the arrangement of the brake at the beginning of the main drive train, the large inertia masses, etc.

Bearings and gears of certain machine tools were damaged because the machine could run when no lubricant was being delivered (faults in the system of interlocking).

Disregard for the considerable dynamic overloads in transient processes was the cause of breakage of gears, shafts and keys in the main drives.

Premature wear of the ways of grinders is observed due to insufficiently reliable protection against abrasive grit.

"*Production causes*" *group*

Casting defects were responsible for breakages of housing-type parts.

Defects of heat treatment led to breakages along the body of gears.

Errors in manufacture and assembly defects led to the breakage of gears, shafts and bearings.

"*Operation causes*" *group*

This group includes:

breakages occurring if speeds are changed when the machine is running at high speed or due to severe braking by engaging reverse rotation;

breakages occurring when the carriage runs up against the headstock or tailstock due to improper adjustment of the stops or because of an oversight of the operator;

breakages due to operation at speeds and/or feeds in excess of the permissible values;

breakages due to the use of the machine tool under conditions not complying with its purpose (for example, in using a general-purpose machine tool as a production model operating under severe cutting conditions);

premature wear of the parts due to poor operation of the lubricating system (irregular changes of oil and cleaning of the system).

If the causes of damages to the machine tools in service tests are established and systematized in this way, measures can be planned for all three groups to improve the dependability and prolong the service life.

Such measures may include: the elimination of direct contact of rubbing surfaces by an extensive application of hydrostatically, air and hydrodynamically lubricated bearings and slideways; reduction in the pressure between mating surfaces; use of antifriction ways and ball-bearing nuts in critical units to replace sliding friction with rolling friction; development of self-aligning multiple-pad sliding-friction bearings that operate dependably (due to the absence of edge pressure) under fluid friction conditions; use of crowned gears and impactless speed changing devices for gears that are frequently shifted in speed gearboxes so as to reduce the most common kind of damage of such gears—wear at the ends of the teeth; application of mechanisms with automatic wear compensation; use of highly efficient lubricating systems, etc.

The quality of the surface layer of a part, its physicochemical properties and microgeometry exert a great influence on the wear resistance and fatigue strength. These two factors are being improved at the present time by an extensive application of surface hardening (flame and induction hardening) and case-hardening (chemical heat treatment) in which the surface layers are saturated with carbon (carburizing), nitrogen (nitriding), chromium (chromizing), both carbon and nitrogen together (cyaniding and carbonitriding), etc.

The part surface can also be hardened by burnishing with rolls, ball or shot peening, electrical-discharge treatment, and metal electroplating (usually chromium plating).

To develop machine tools of high-class design, it is necessary to have, in addition to other important data, information on the loads to which the machines are subjected in regular service: their magnitude and how they vary with time. A knowledge of the actual loads, especially in the case of general-purpose machine tools for which variable duty is typical, enables dependable constructions to be developed, with no excess margin of safety that makes the machine heavier and increases its overall size.

A study of actual loads, based on service observations, has been carried out by ENIMS beginning with 1957. During a shorter or longer period (from several hours to several weeks), machine tools of the widely used models were equipped with special recording instruments which registered the total time of operation, in performing regular jobs, in each interval of motor power consumption values and spindle speeds.

The torque values were calculated from the power consumption and corresponding spindle speeds. A study of the generalized data, obtained for various models of machine tools operating under various service conditions, concerning the distribution of operating time at various values of the avail-

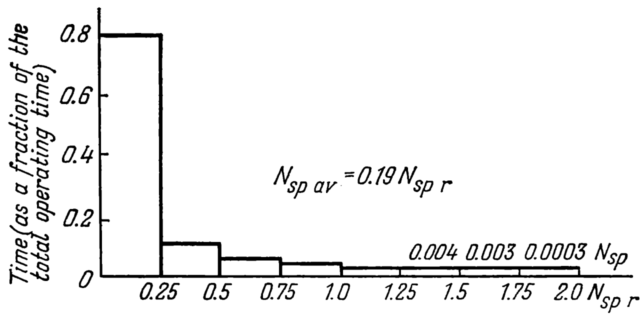


Fig. 346. Graph showing the distribution of operating time at various values of power developed on the spindles of general-purpose lathes and milling machines (generalized statistical data)

able power on the spindles of general-purpose lathes and milling machines, showed that during about 80 per cent of their operation time they utilize less than one-fourth of the rated power N_r, during about 90 per cent they use less than one half of N_r and during only 1 or 2 per cent do they work at values exceeding N_r (Fig. 346). The average value of the utilized power in respect to time is (0.18 to 0.2) N_r.

The utilization of the speed capacity of machine tools is characterized by the ratio $\frac{n_{av}}{n_{max}}$. According to the results of service observations, this ratio reaches 0.45 to 0.6 for lathes, models 1Д62M, 1Д63 and 1A62; 0.2 to 0.25 for the model 1K62 lathe, and is only 0.15 to 0.2 for milling machines, models 6H12, 6H82, 6H13 and 6H83. Average generalized data on the distribution of torques in respect to the time of operation are similar to the data concerning power utilization. The maximum value of the torques registered on the spindles of lathes and milling machines of the above-mentioned models was approximately 50 kgf-m, which for lathes was only from 40 to 67 per cent of the maximum value listed in the machine certificate as permitted by the strength of the weakest member of the drive train.

Graphs of the type shown in Fig. 346 were treated by means of correlation equations to obtain the smooth law representing the statistic dependence of the time or number of loading cycles on the load. Curves of two correlation equations for two groups of lathes are illustrated in Fig. 347.

The results of this study of the service load conditions of general-purpose machine tools present very valuable data since they enable the accepted values of design loads to be refined. A knowledge of the service load conditions is also necessary in experimentally investigating the performance of machine tool parts.

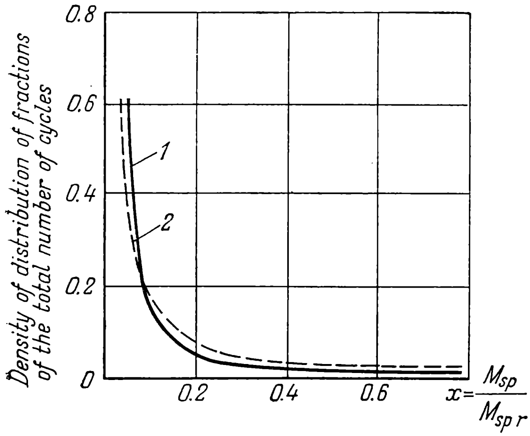


Fig. 347. Curves of correlation equations:

1—for lathes, according to generalized data $\left(y=\frac{0.019}{x^2}\right)$;

2—lathes engaged in roughing operations $y=\left(\frac{0.071}{x^{3/2}}\right)$

As mentioned earlier, one of the reasons for the failure of the components of machine tool drives is their breakage from cyclic loads due to the development of fatigue cracks. Though phenomena occurring in the system of the drive are quite complex, the dependability and durability of the components can be sufficiently accurately and rapidly assessed if short-term bench tests are conducted together with the calculations and comparatively prolonged service observations.

The technique of such tests was worked out by R. Pratusevich of ENIMS. To ensure that the operating conditions of the parts approximate the actual production conditions, the machine tool drive was tested in the assembled condition according to a loading programme (for machine tools produced in large lots). The endurance bench tests of the drive were curtailed by forcing the magnitude of the load and not the frequency of its application (this is due to the relationship between the magnitude of the load Q and the number of cycles N_{cy} of its action required for failure: $Q^m N_{cy} = \text{const}$, where $m \geqslant 3$).

Load increases were assigned in such a manner that the maximum stress developed in the weakest members of the train did not exceed the yield point of the material, i.e., about 2 or 3 times the design (nominal) stress.

The loading programme was worked out in accordance with the data on the service load conditions of general-purpose machine tools. In particular, in respect to the speed gearboxes of general-purpose medium-size lathes and milling machines, this schedule was drawn up in accordance with the equation $y=\frac{c}{x^{3/2}}$ characterizing load conditions for severe duty (see Fig. 347). The length of the period of load variation was taken as approximately 5 minutes (Fig. 348). The speed conditions in the tests were also assigned in accordance with the data of the service observations so that a large number of speed changes took place in the gearbox before failure of the weakest member.

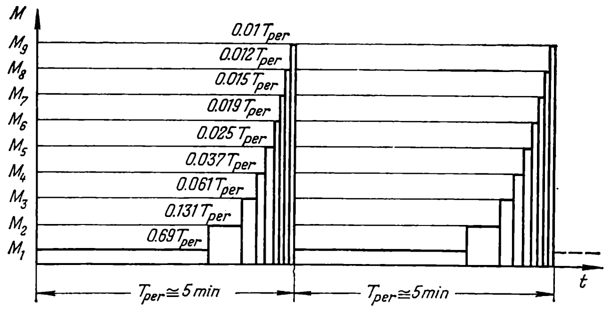


Fig. 348. Programme of periodic load variation on the input shaft of the test bench

The speed gearboxes of general-purpose machine tools were tested with forced loads by the procedure described above in a test bench developed in ENIMS. This bench is based on a scheme with an open-loop power circuit (Fig. 349). The test bench consists of induction motor *1* which drives two identical lot-produced speed gearboxes *3* and *5* through V-belts *2*. The gearboxes are linked together by coupling *4*. Generator *6* at the output end of the test bench develops the brake torque (the torque value is selected so that the load on the weakest member of the drive train is 2 or 3 times the nominal value under regular operating conditions). When the same combinations of gearing are engaged in the two gearboxes the speed of rotation of the generator shaft remains constant for any speed of the spindles. The torque is varied according to the given test programme by changing the current in steps in the generator field circuit. This is done by means of master switch *7* (type КЭП-12УТ, made by the Phyzpribor Plant) which is driven by an induction motor. Also mounted on the bench were a number of instruments for checking the magnitude of the load and the nature of its variation with time, the time of bench operation up to the failure of any part and the number of loading cycles.

These bench tests revealed factors concerned with manufacture and design that reduced the service life of various components of the drive.

The results of bench tests, substantiated by selected data from machine tool service practice, enabled defects to be revealed in the heat treatment

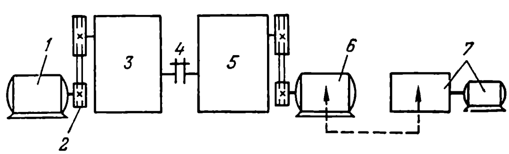


Fig. 349. Schematic diagram of a test bench with programmed loading

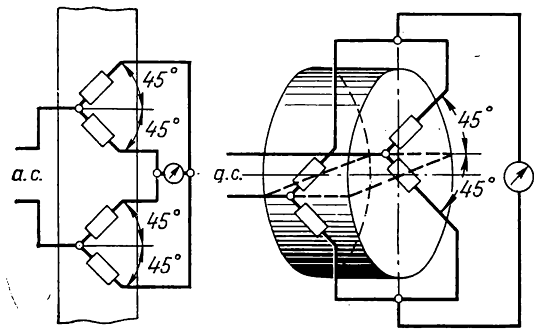


Fig. 350. Arrangement of wire-type pickups on a shaft for measuring the torque

of the tooth spaces and tooth flanks. These defects considerably reduced the fatigue strength and, consequently, the service life of many gears. It is highly desirable to carry out such tests on the pilot models of machine tools that are to be put into lot production.

Machine tool practice and the service tests have shown that the breakage of components of the main drive is often due to high dynamic loads to which the drive is subject in the cutting process. Of interest in this connection are investigations of vibration phenomena in machine tool drives. The work of E. Rivin of ENIMS can be cited as an example. The experimental part of this work was done on a knee-type milling machine with a table 1,600 × × 400 mm in size. The loads acting in the milling process were recorded in oscillograms which were made with the aid of wire-type pickups glued on five intermediate shafts of the speed gearbox in this machine according to the scheme for measuring the torque (Fig. 350). The pickups were arranged on several shafts of the gearbox so as to follow the development of the vibration processes along the train and to obtain more reliable and complete data on the dynamic loads. In the model 1K62 lathe, the pickups were glued on the friction clutch shaft (input shaft of the speed gearbox), on the spindle and on the shaft directly preceding the spindle. The leads from the pickups were brought out through axial holes drilled in the shaft ends, and were connected to the current collector.

The signals of the pickups were supplied through current collecting devices linked by a flexible shaft to the end of the shaft being investigated (Fig. 351).

The oscillograms showed, in particular, that the torque developed on the intermediate shafts of the drive varies significantly during the milling process, the nonuniformity factor being

$$\frac{M_{max}}{M_{av}} = 1.5 \text{ to } 3.2$$

In another work by E. Rivin and others, the same factor is given as $\frac{M_{max}}{M_{av}} = 1.1$ to 2 for rough and finish turning a plain steel ingot in a lathe. These examples prove that the prevailing opinion on the absence of appreciable oscillatory loads in the components of the drive during smooth cutting is wrong.

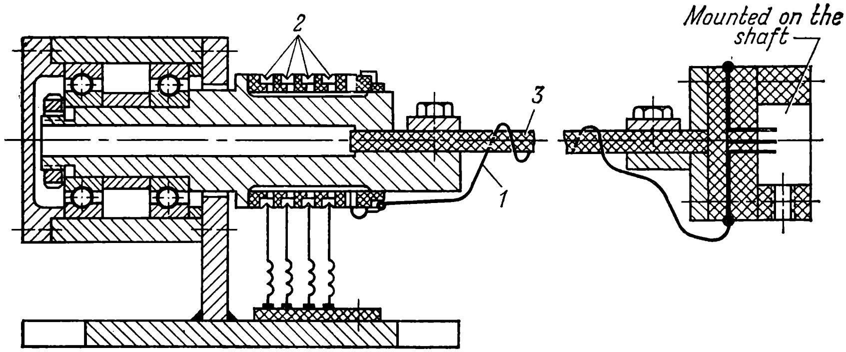


Fig. 351. Current collecting device with a flexible connecting shaft (developed in ENIMS):
1—lead; 2—collector rings; 3—flexible shaft

The magnitude of the nonuniformity factor depends upon the design features of the machine tool drive, i.e., its dynamic characteristics, such as natural frequencies, form of vibration, damping, kind of machining being done, speeds and feeds, cutting tools used, etc.

It stands to reason that a neglect of this large variation in torque in designing a machine tool may lead to a marked reduction in the service life. For this reason, it is expedient to conduct experimental and theoretical investigations on the dynamic loads for all machine tools manufactured in large lots.

The investigations carried out in ENIMS on several machine tools showed that it is possible to reduce the amplitude of dynamic loads by 30 to 40 per cent by rationally constructing the structural diagram and making dynamic calculations of the drive. This enables the dependability of a machine tool drive to be considerably improved and its service life to be prolonged.

Along with measures concerned with the design and manufacture, rational use of machine tools is of great importance for improving their dependability and durability. Training of the machine operator, responsibility for the maintenance, proper operation and care of the equipment, as well as the implementing of all measures scheduled in the accepted system of preventive repairs are all necessary to ensure the durability of the machine tool that is inherent in its construction.

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